OPTIMIZATION OF AMMONIA REFRIGERATION SYSTEM

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Certificate

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DEDICATIONS

We would like to dedicate this thesis to the endless support of our parents, teachers and to our hard work as a team.

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We would like to thank our supervisor Dr. Sarah Farrukh for her countless efforts and continuous guidance on our project. We also extend our thanks to Mirza Muhammad Nabeel and Engro Foods for providing us with complete guidance throughout our project.

ABSTRACT

Refrigeration systems are majorly used in food industry especially in manufacturing of dairy products. “GEA Grasso Ice-water system” is one of these refrigeration systems being used at Engro foods. With every running plant there is always an area for improvement to make it more energy and cost efficient. With our solutions to enhance the COP of the current ammonia refrigeration system, we have managed to make it cost effective. It includes using R32 as a potential refrigerant and VFDs (Variable Frequency Drives) on both the condenser and compressor fans. Design considerations for both the Evaporator and Condenser are also discussed

ABBREVIATIONS

Sr #	Symbol/Abbreviation	Explanation
1	Q	Heat Transfer rate
2	$\dot{m}$	Mass flow rate
3	H_{out}	Enthalpy at outlet
4	H_{in}	Enthalpy at inlet
5	v_o	Velocity at outlet
6	v_i	Velocity at inlet
7	Q_{in}	Heat in
8	W_{shaft}	Shaft work
9	η	Efficiency
10	$\Delta Hisen$	Isentropic Enthalpy change
11	$\Delta Hact$	Actual Enthalpy change
12	Cp	Specific heat Capacity
13	ΔT	Temperature difference
14	Q_H	Heat rejected by condenser
15	T_{ref}	Reference Temperature
16	λ	Latent heat of vaporization
17	x	Vapour fraction
18	H_v	Enthalpy of vapour
19	H_l	Enthalpy of liquid
20	$\dot{W}_s$	Shaft work
21	kJ	Kilo Joules
22	kW	Kilo watt
23	kWh	Kilo watt hour
24	kg	Kilo gram
25	COP_R	Refrigeration Coefficient Of Performance
26	kPa	Kilo Pascal
27	°C	Degree Celsius
28	T_i	Inlet Temperature
29	T_o	Outlet Temperature
30	Q_{evap}	Cooling Load/Heat transfer rate of evaporator
31	ε	Effectiveness
32	C_h	Heat capacity rate of hot stream
33	C_c	Heat capacity rate of cold stream
34	C	Heat capacity rate ratio
35	NTU	Number of Transfer Units
36	Ts	Surface Temperature
37	U	Overall heat transfer coefficient
38	$\Delta\chi$	Thickness of tube/ Difference in inner and outer diameter
39	W	Watt
40	h_{water}	Heat transfer coefficient of water
41	$h_{ammonia}$	Het transfer coefficient of ammonia
42	Do	Outside diameter
43	Di	Inside diameter
44	kw	Conductivity of tube wall
45	k_{copper}	Conductivity of copper
46	As	Surface Area

47	L	Length of tube
48	$\bar{\wedge}$	Pi (mathematical constant with a value of 3.14)
49	n	Number of passes or tubes
50	ΔP	Pressure drop
51	N_{Re}	Reynold’s number
52	μ	Viscosity of fluid
53	v	Velocity of fluid
54	f	Darcy friction factor
55	ρ	Density of fluid
56	Pr	Prandtl number
57	S	Entropy
58	Relative H	Relative Humidity
59	P_i	Inside Pressure
60	τ	Tensile stress
61	**T**	Thickness of tube
62	A_i	Inside Area
63	h_i	Inside heat transfer coefficient
64	Nu	Nusset number
65	$\frac{\gamma}{d_t}$	Mass flow of water film per meter square of tube
66	LMTD	Log Mean Temperature Difference
67	$AMTD$	Average Mean Temperature Difference
68	P-H	Pressure Enthalpy
69	T-S	Temperature-Entropy Diagram
70	ODP	Ozone Depletion Potential
71	GWP	Global Warming Potential
72	R-717	Refrigerant-717(Ammonia)
73	R-32	Refrigerant-32 (Diflouromethane)
74	Psi	Pound per square inch
75	EPA	Environmental Protection Agency
76	PSM	Process Safety Management
77	CFC	Chloro Flouro Carbon

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Chapter 1 INTRODUCTION

1.1 THE GENERIC REFRIGERATION CYCLE:

A refrigeration cycle is very important when it comes to storing and cooling in both the food industry and at homes. The most common refrigerant used in the industry is ammonia because of its unique thermodynamic properties explained later. It boils at -27 degrees F. This is what happens to keep the refrigerator cool:

1. The compressor compresses the ammonia gas. The compressed gas heats up as it is pressurized.
2. The coils on the back of the refrigerator let the hot ammonia gas dissipate its heat. The ammonia gas condenses into ammonia liquid at high pressure.
3. The high-pressure ammonia liquid flows through the expansion valve. You can think of the expansion valve as a small hole. On one side of the hole is high-pressure ammonia liquid. On the other side of the hole is a low-pressure area (because the compressor is sucking gas out of that side).
4. The liquid ammonia immediately boils and vaporizes, its temperature dropping to -27 F. This makes the inside of the refrigerator cold.
5. The cold ammonia gas is sucked up by the compressor, and the cycle repeats

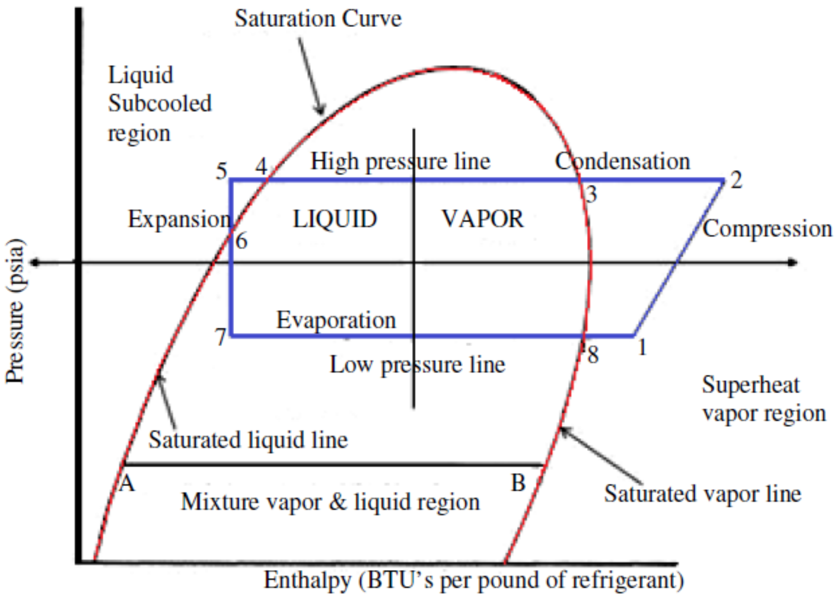


Figure 1 GENERIC REFRIGERATION CYLCE DIAGRAM

In our project we had to optimise the current refrigeration system in Engro's plant that is used to cool the process water. The average capacity of the plant is 4,838,400 kgs per day of process water cooled. The following describes a simple cycle with the relevant process conditions

- Ammonia at 3.84 bar and -3℃ goes to the screw compressor where it is compressed to 14 bar and 97.3 ℃.
- This ammonia vapor is cooled in the evaporative condenser to 36℃ and is stored in the High Pressure Tank.
- After the throttling valve (globe valve) throttles ammonia to -3 by the Joule Thomson effect, the partially vaporized ammonia goes to the Low pressure Tank.
- This partially vaporized liquid ammonia goes to the Ice Bank evaporator and leaves it at the same temperature it went in. By heat transfer through the evaporator tubes, ammonia is completely vaporized.
- This ammonia vapor then goes to the compressor and the cycle repeats again. The Process water enters at 8 ℃ and leaves at 0℃.

In order to optimise the current system we identified three key areas; improving the evaporator design, improving the condenser design and using another refrigerant other than Ammonia.

Evaporator:

- By increasing the surface area of contact between the water and ammonia.
- Decreasing the water flow rate.
- Increasing the ammonia flow rate.

Condenser:

- Changing the water and air flow rates in the evaporative condenser
- Increasing the surface area by choosing a specific tube arrangement.
- Using VFDs on the air motor.

Refrigerant:

- Non-CFC refrigerants were studied and R-32 was found as a potential refrigerant.
- By using R-32 (Difluoromethane), smaller compressors can be used and the refrigeration capacity increases by 60%.
- It reduces the long term operating costs of the refrigeration system.

Chapter 2 LITERATURE REVIEW

Industrial refrigeration systems based on vapour compression cycle using ammonia as a refrigerant usually have four basic units. The units include:

1. Compressor
2. Condenser
3. Throttling valve
4. Evaporator

other than these four basic units the refrigeration systems may also have tanks, thermocouples, pressure gauges and pumps.

Thermodynamic cycles can be categorized into gas cycles and vapour cycles. In a typical gas cycle, the working fluid (a gas) does not undergo phase change, consequently the operating cycle will be away from the vapour dome. In gas cycles, heat rejection and refrigeration take place as the gas undergoes sensible cooling and heating. In a vapour cycle the working fluid undergoes phase change and refrigeration effect is due to the vaporization of refrigerant liquid. If the refrigerant is a pure substance then its temperature remains constant during the phase change processes. However, if a zeotropic mixture is used as a refrigerant, then there will be a temperature glide during vaporization and condensation.

Since the refrigeration effect is produced during phase change, large amount of heat (latent heat) can be transferred per kilogram of refrigerant at a near constant temperature. Hence, the required mass flow rates for a given refrigeration capacity will be much smaller compared to a gas cycle. Vapour cycles can be subdivided into vapour compression systems, vapour absorption systems, vapour jet systems etc. Among these the vapour compression refrigeration systems are predominant.

Industrial refrigeration systems typically use ammonia (R-717) as a refrigerant and can have single stage or multiple stage compression. Different evaporator configurations are common such as: Direct Expansion (DX), Flooded and Liquid Overfeed and a hybrid combination of the above configurations and are based on the process requirements. Secondary coolants and economized compression also factor in different design configurations.

All pumped liquid overfeed systems require receiver vessels that hold two phase refrigerant. Based on the actual design requirements, a refrigeration plant may have several receiver tanks. The first is the high pressure receiver where liquid refrigerant exiting the condenser is stored. Liquid refrigerant from the high-pressure receiver is then throttled either to the intermediate pressure receiver or to the direct expansion evaporators in the 45-50 °F cooling load rooms. The backpressure regulator throttles the refrigerant gas to the intermediate pressure receiver, which is at a lower temperature/pressure. Liquid in the intermediate pressure receiver is then either pumped to the 25-30 °F cooler or throttled again to the low-pressure receiver. Liquid refrigerant from the low pressure receiver is pumped to the –10 to –15 °F freezer loads with a mechanical liquid recirculating pump.

The factors that influence the refrigeration system energy use are the inherent efficiency of the design and refrigerant, the condition of the equipment, the control strategy, and the load profile of the system (deviation of the operating cooling loads from the design cooling loads). In many cases, the installed refrigeration plants can benefit from an optimization process that incorporates monitoring of key operating parameters resulting in subsequent control adjustments and or system operational changes based on the assessed data

Typical ammonia refrigeration system energy savings strategies are presented below:

1. Reduce heat loads (Low Cost Strategies)
2. Turn off lights in unoccupied refrigerated spaces and use more efficient lights
3. Increase insulation
4. Reduce infiltration
5. Check all coils for dirt and debris or missing nozzles in condensers
6. Reduce temperature lifts in the refrigeration plant
7. Use efficient compressors and operate them optimally (sequencing)
8. Ensure that defrost controls are set to optimize defrost effectiveness
9. Use optimum size of evaporators and condensers – should be sized to operate at lowest condensing temperature and highest effective evaporating temperature
10. Install VFDs on evaporators, condensers and compressors
11. Use premium efficiency motors on compressors, condensers, evaporators and pumps
12. Computer controls – facilities improved and optimized operating strategies
13. Recover heat from at or before the Condenser for process or domestic hot water usage

2.1 Compressor:

Compressors are the major source of consuming energy in a refrigeration system and that is the reason that its selection and operation are critical in determining the efficiency of the cycle. The type of compressor, conditions at suction and discharge along with unloading controls effect the efficiency of compressor as well as the cycle. Compressors that are used in industrial refrigeration systems can be of different types but mainly following types of compressors are used based on required specification of the cycle:

- Reciprocating
- Screw
- Rotary vane designs
- Variable speed centrifugal compressors.

These compressors can be further classified as

- Open Drive Or Hermetic
- Single stage Or multi stage compressor

2.2 Reciprocating versus Screw Compressor:

The comparison of these compressors is based on following

2.2.1 Operating Range:

Reciprocating compressors are used when the operating range of compressor is below 75 kW and Screw compressors are used when the range is above 75kW.

2.2.2 Load Capacity:

Screw compressors have shown high efficiency while operating at full-load capacity but same compressors have shown least efficiency when operating below the 50% capacity when compared with reciprocating machines.

2.2.3 Load Variability:

Load variability is the variation in the capacities of the compressor. Reciprocating compressors have shown good result with the systems that have high load variability.

2.2.4 Load Sharing:

Load sharing refers to the use of multiple compressors to meet the required specs in terms of load.

Since screw compressor show least efficiency below 50% of full-load, so using screw compressors below 50% load is not recommended.

For medium loads, the small compressor should be fully loaded.

For higher loads, the larger compressor should be fully loaded.
While using reciprocating and screw compressors together, the screw compressor should be base loaded and reciprocating compressor should be setup to meet the varying load.

2.3 Open Drive versus Hermetic Compressors:

Compressors are classified as open drive and hermetic on the basis of location of motor. In open drive compressors the shaft is rotated by the motor located outside the shell assemble while in hermetic compressors the motor and the the compressor assembly is sealed in the same welded shell.
Usually Open drive compressors are preferred over hermetic because the chances of heat losses are more in hermetic compressors due to their compact design.

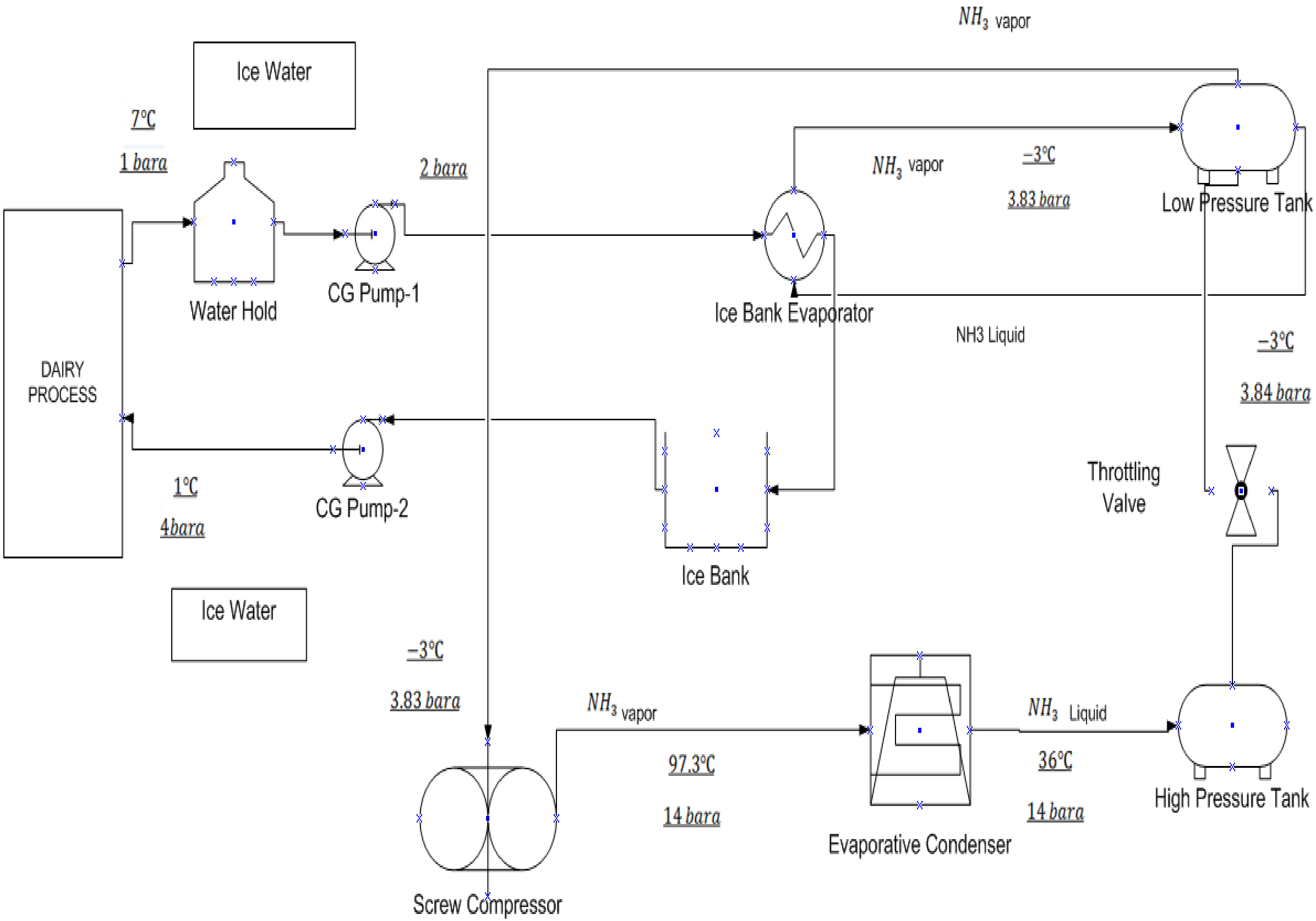
2.3.1 One Stage V/s Two Stage Compression

Single-stage compression is most commonly used in single temperature applications with moderate to high suction temperatures. As the temperature requirements fall below –15 □F, the pressure ratios increase at which the single-staged ammonia systems have to operate at higher discharge temperatures, which reduces the compressor efficiency.

Two-stage compression systems are a possible alternative when high-pressure lift (pressure difference between suction and discharge pressures) is required such as in low temperature applications (below 0 °F for ammonia). By using two or more stages of compression, sub-cooling the refrigerant at each stage of compression increases the overall system operating efficiency. Multi-staging is also ideal for applications that require different temperatures. In multi-stage systems, flash-type intercoolers are more efficient than shell-and coil intercoolers.

Where either single or two stage compression systems can be used, two stage systems require less power and have lower operating costs, but can have a higher initial equipment cost.

Chapter 3 PROCESS FLOW DIAGRAM



NH_3 vapor
Ice Water
7℃
1 bara
2 bara
NH_3 vapor
−3℃
3.83 bara
Low Pressure Tank
Water Hold
CG Pump-1
Ice Bank Evaporator
NH3 Liquid
DAIRY PROCESS
−3℃
3.84 bara
Throttling Valve
1℃
4bara
CG Pump-2
Ice Bank
Ice Water
−3℃
3.83 bara
NH_3 vapor
97.3℃
14 bara
NH_3 Liquid
36℃
14 bara
High Pressure Tank
Evaporative Condenser
Screw Compressor

Chapter 4 MATERIAL BALANCE

4.1 BASED ON CURRENTLY WORKING PLANT CONDITIONS:

Material Balance to calculate ammonia and water flow rate

Where:

Cooling load is in kW

H is in kJ/kg

C_p is in kJ/kgK

ΔT is in K

$\dot{m}$ is in kg/s

4.1.1 <u>Water Flow rate</u>

Equation of Sensible heat:

$$Q = \dot{m}C_p\Delta T$$

$$\dot{m} = Q/C_p\Delta T$$

$$\dot{m} = \frac{1408}{4.18 * 6}$$

$$\dot{m} = 56.14\ kg/s$$

4.1.2 <u>Ammonia flow rate</u>

H1=1458.326

H4=1633.288

ΔH=175

Cooling load=1408 kW

Cooling provided per kg of ammonia=1085.869

ṁ=Cooling load/Cooling provided per kg

=1408/1085.869

=1.296 kg/s

Chapter 5 ENERGY BALANCE

5.1 Compressor

Assumptions:

- Steady flow inside compressor and uniform condition at inlet and outlet
- No change in elevation
- No KE and velocity change

$$\dot{m}\left(H_{out} - H_{in} + \left(\frac{(v_o)^2 - (v_i)^2}{2}\right) + g(z_o - z_i)\right) = Q_{in} + W_{shaft}$$

$$\dot{m}(H_{out} - H_{in}) = Q_{out} + W_{shaft}$$

m =1.296kg/s

H_{out} =1633.288

H_{in} = 1458.326

$$W_{shaft} = 350\ kW$$

1.296(1633.288-1458.326)=Qout+350

Q_{out}=-123.13 kW

The Efficiency of the compressor is:

$\eta = (\dot{m} * (H_{out} - H_{in}) / \dot{m}C_p\Delta T$ Cp=2.596+2.630/2=2.613

$\eta = (1.296(1633.288 - 1458.326)) / (1.296) * (2.613) * (97.3 + 3)$

$\eta = 0.67$

The total electrical energy doesn't get converted to shaft work hence the value for efficiency.

5.2 Evaporative Condenser

Latent heat of vaporization of ammonia (NH_3) = 1369 kJ/kg

So the total heat rejected (Q_H) = ($\dot{m} * 1369$) + $\dot{m}C_p\Delta T$

5.2.1 To calculate this duty:

$$ElectricalLoad = CompressorLoad + FanMotorLoad$$

$$= (350 - 123.11) + 37 = 264\,KW$$

because the heat was not converted to work and wasted

Heat Addition from the compressor (Q) = Cooling capacity + electrical load

$= 1408\,kW + 264\,kW$

$(Q) = 1672kW$

In lieu of the Energy Balance,

Heat addition (Q) = Heat rejected by the condenser(Q_H) :

Calculating(Q_H):

- Taking the value of $C_{p,}$as
  C_p=(4.885+2.596)/2=3.740
- Hence $Q \approx (Q_H)$
  (Q_H) = $(\dot{m} * 1369) + \dot{m}C_p\Delta T$

(Q_H) = $(1.296 * 1369) + \big(1.296 * 3.740 * (97.3 - 36)\big)$

(Q_H) = $1758\ kW$

5.3 Throttling Valve

As NH_3 liquid is throttled through the globe valve some of it turns to vapor.

To calculate the amount that turns to vapour

Taking $T_{ref} = 25$

Q required to heat the vapours:

$Q = (\dot{m}C_p\Delta T)_{in} - (\dot{m}C_p\Delta T)_{out}$

$Q = \dot{m}[(C_p\Delta T)_{in} - \big(C_p\Delta T\big)_{out}]$

Cp=(4.885+2.655)/2

$$Q = 1.29(3.770 * (36 - 25))_{in} - (3.770 * (-3 - 25)_{out}]$$

Q=ṁ*C_p*ΔT

Q=1.296*3.770*39

Q=190.551 kJ/s

m=Q/ λ

m=190.55/1369

m=0.140 kg/s

According to the balance:

Initial Enthalpy = Enthalpy of NH_3 liquid + Vapor Enthalpy

H_{in} = 352 kJ/kg

H_{out} = $(H_v * x) + (H_l * (1 - x))$

H_{out} = $(1438.76 * 0.14) + (167 * (1 - 0.14))$

$$H_{out} = 348 \frac{kJ}{kg}$$

$$H_{out} \approx H_{in}$$

5.4 Ice Bank Evaporator

The heat released from the 7°C process water is taken as latent heat by the ammonia which converts from liquid to vapors.

$$Q = \dot{m} C_p \Delta T$$

$$Q = 56.1 * 4.18 * (7 - 1)$$

$$Q = 1408 \frac{kJ}{s}$$

The latent heat required to change liquid ammonia to vapor state is the latent heat of vaporization of the value 1369 kJ/kg. So this released heat is sufficient enough for phase change for ammonia but not for a temperature change.

5.5 Pump

Using the first law

$$\dot{m}(H_2 - H_1) = \dot{W}_s + \dot{Q}$$

But no work is done on or by the pump so $\dot{Q} = 0$

The equation reduces to

$$(H_2 - H_1) = \dot{W}_s / \dot{m}$$

Water flow rate = 56.14 kg/s

1 kg/s = 3.6 m3/hr

Volumetric flow rate of water = 56.14*3.6

Volume (V) of the Pump = 0.056 m3/s

After the derivation we get

$\Delta H = (H_2 - H_1) = (V\ (P_2 - P_1))\dot{m}$

$\Delta H = (H_2 - H_1) = (0.056 * (200 - 100))/56.14$

$$\Delta H = 0.100 \frac{kJ}{kg}$$

From the H values:

$\Delta H = (H_2 - H_1)$

$\Delta H = 29.6 - 29.5$

$$\Delta H = 0.100 \frac{kJ}{kg}$$

To calculate COP_R

$COP_R = Q_e/W_s$

Where Q_e is the system cooling load and $W_s(kW)$is the work done by the compressor.

Q_e = 1408 kJ/s

$$COP_R = \frac{1408}{(350)}$$

$COP_R\,(\omega) = 4$

Chapter 6 NEW MATERIAL BALANCE

6.1 BASED ON NEW DESIGN CONDITIONS

Material Balance to calculate ammonia and water flow rate

Where:

Cooling load is in kW

H is in kJ/kg

C_p is in kJ/kgK

ΔT is in K

$\dot{m}$ is in kg/s

Water Flow rate

Equation of Sensible heat:

$$Q = \dot{m} C_p \Delta T$$

$$\dot{m} = Q/C_p \Delta T$$

$$\dot{m} = \frac{1408}{4.18 * 8}$$

$$\dot{m} = 42.105\ kg/s$$

Ammonia flow rate

Given:

Cooling load*=1408kW

Cooling provided per kg of refrigerant*=1100 kJ/kg

$$\dot{m} = \frac{Cooling\ load}{Cooling\ provided\ per\ kg\ of\ Ammonia}$$

=1408/1100

=1.280 kg/s

Chapter 7

NEW ENERGY BALANCE

7.1 Compressor

Assumptions:

- Steady flow inside compressor and uniform condition at inlet and outlet
- No change in elevation
- No KE and velocity change

$$\dot{m}\left(H_{out} - H_{in} + \left(\frac{(v_o)^2 - (v_i)^2}{2}\right) + g(z_o - z_i)\right) = Q_{in} + W_{shaft}$$

$$\dot{m}(H_{out} - H_{in}) = Q_{out} + W_{shaft}$$

m =1.280kg/s

H_{out} =1633.288 kJ/kg

H_{in} =1465.700kJ/kg

$W_{shaft} = 302.000\ kW$

1.280(1633.288-1465.700)= $\mathbf{Q_{out}}$+302

$\mathbf{Q_{out}}$=-87.487 kW

The Efficiency of the compressor is :

$\eta = (\dot{m} * (H_{out} - H_{in})\ /\dot{m}C_p\Delta T$

Cp=2.590+2.596/2=2.593

$\eta = (1.280(1633.2881465.700))\ /\ (1.280) * (2.593) * (97.300 - 0)$

$\eta = 0.670$

The total electrical energy doesn't get converted to shaft work hence the value for efficiency.

7.2 Evaporative Condenser

Latent heat of vaporization of ammonia (NH_3) = 1369 kJ/kg

So the total heat rejected (Q_H) = ($\dot{m} * 1369$) + $\dot{m}C_p\Delta T$

<u>To calculate this duty:</u>

$$Electrical\ Load = Compressor\ Load + Fan\ motor\ load$$

$$= (302 - 87.487) + 48.699 = 263.212\ kW$$

because the heat was not converted to work and wasted

Heat Addition from the compressor (Q) = Cooling capacity + electrical load

$= 1408\ kW + 263.212\ kW$

$(Q) = 1671.000kW$

In lieu of the Energy Balance,

Heat addition (Q) = Heat rejected by the condenser(Q_H) :

Calculating(Q_H):

Taking the value of $C_{p,out}$as

Cp=(4.865+2.596)/2=3.731

(Q_H) = $(\dot{m} * 1369\ ) + \dot{m}C_p\Delta T$

(Q_H) = $(1.280 * 1369\ ) + (1.280 * 3.731 * (97.3 - 34))$

(Q_H) = $1670kW$

Hence $Q \approx (Q_H)$

7.3 Throttling Valve

As NH_3 liquid is throttled through the globe valve some of it turns to vapor.

To calculate the amount that turns to vapor

Taking $T_{ref} = 25$

Q required to heat the vapors:

$Q = (\dot{m}C_p\Delta T)_{in} - (\dot{m}C_p\Delta T)_{out}$

$Q = \dot{m}[(C_p\Delta T)_{in} - (C_p\Delta T)_{out}]$

Cp=(4.865+2.622)/2

$Cp = 3.744$

$$Q = 1.280[(3.744 * (34 - 25))_{in} - (3.744 * (0 - 25)_{out}]$$

Q=ṁ*Cp*ΔT

Q=1.280*3.744*34

Q=165.376 kJ/s

ṁ =Q/ λ

ṁ =165.376/1369

ṁ =0.128 kg/s

According to the balance:

Initial Enthalpy = Enthalpy of NH_3 liquid + Vapor Enthalpy

H_{in} = 361.233 kJ/kg

H_{out} $= (H_v * x) + (H_l * (1 - x))$

H_{out} $= (1460.500 * 0.128) + (190 * (1 - 0.128))$

$$H_{out} = 355.100 \frac{kJ}{kg}$$

$$H_{out} \approx H_{in}$$

7.4 Ice Bank Evaporator

The heat released from the 8°C process water is taken as latent heat by the ammonia which converts from liquid to vapors.

$$Q = \dot{m} C_p \Delta T$$

$$Q = 42.105 * 4.18 * (8 - 0)$$

$$Q = 1408 \frac{kJ}{s}$$

The latent heat required to change liquid ammonia to vapor state is the latent heat of vaporization of the value 1369 kJ/kg. So this released heat is sufficient enough for phase change for ammonia but not for a temperature change.

7.5 Pump

Using the first law

$\dot{m}(H_2 - H_1) = \dot{W}_s + \dot{Q}$

But no work is done on or by the pump so $\dot{Q} = 0$

The equation reduces to

$(H_2 - H_1) = \dot{W}_s /\dot{m}$

Water flow rate = 42.105 kg/s

1 kg/s = 3.6 m3/hr

Volumetric flow rate of water = 42.105*3.6

Volume (V) of the Pump = 0.056 m3/s

After the derivation we get

$\Delta H = (H_2 - H_1) = (V\ (P_2 - P_1))\dot{m}$

$\Delta H = (H_2 - H_1) = (0.056 * (200 - 100))/42.105$

$\Delta H = 0.133 \frac{kJ}{kg}$

From the H values:

$\Delta H = (H_2 - H_1)$

$\Delta H = 33.822 - 33.623$

$\Delta H = 0.199 \frac{kJ}{kg}$

<u>To calculate COP_R</u>

$COP_R = Q_e/W_s$

Where Q_C is the system cooling load and$W_s(kW)$is the work done by the compressor.

$Q_C = 1408$ kJ/s

$COP_R = \frac{1408}{(302)}$

$COP_R\,(\omega) = 4.622$

The following is a chart that enlists the Condesing Temperature and the Wet Bulb temperature availaible that collectively gives the Correction factors. To find out the condenser load, wet bulb temperature plays a very important.

If the outside wet bulb temperature is high, the power of the condesner has to be high in order to reject the heat. For choosing the condenser compatible with the wet bulb temperature, the correction factor has to be around 1.

So for our project the condesing temperature was 36 and the corresponding wet bulb temperature was 21.

Ammonia Heat Rejection Factors / Capacity Factor																					
Condensing Pressure		Condensing Temperature	Wet Bulb Temperature																		
Kpa	Bar	C	C																		assumed
			10	12	14	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	32
1063	10.63	30	0.95	1.03	1.12	1.23	1.31	1.40	1.31	1.63	1.79	1.99	2.24	2.56	3.00						
1133	11.33	32	0.84	0.90	0.97	1.06	1.12	1.18	1.25	1.32	1.43	1.55	1.70	1.88	2.11						
1206	12.06	34	0.76	0.81	0.86	0.93	0.98	1.02	1.07	1.12	1.19	1.28	1.38	1.48	1.61	1.80	2.06				
1245	12.45	35	0.71	0.76	0.81	0.87	0.91	0.95	0.99	1.03	1.08	1.15	1.23	1.30	1.39	1.53	1.69	1.90	2.15	2.47	3.32
1284	12.84	36	0.69	0.73	0.77	0.82	0.86	0.89	0.92	0.96	1.01	1.07	1.13	1.20	1.28	1.39	1.53	1.70	1.91	2.17	2.84
1365	13.65	38	0.63	0.66	0.69	0.73	0.76	0.78	0.81	0.83	0.86	0.90	0.94	0.99	1.05	1.12	1.21	1.31	1.44	1.59	1.95
1451	14.51	40	0.58	0.60	0.62	0.65	0.67	0.70	0.72	0.74	0.76	0.80	0.83	0.87	0.91	0.96	1.02	1.09	1.18	1.29	1.57
1539	15.39	42	0.53	0.55	0.57	0.60	0.61	0.63	0.64	0.66	0.68	0.71	0.74	0.76	0.80	0.84	0.88	0.93	0.99	1.06	1.21
1630	16.3	44	0.49	0.50	0.52	0.54	0.56	0.56	0.58	0.59	0.61	0.63	0.65	0.67	0.70	0.73	0.76	0.79	0.83	0.86	0.91

Chapter 8 DESIGN SPECIFICATIONS

8.1 ICE BANK EVAPORATOR

There are several evaporators used in the industry. Engro uses the Ice Bank Evaporator which is a tank made of civil structure and the coils are made of SS-316.This evaporator is an old fashioned design that is still used in the industry because of its simple style and the ability to achieve the desired conditions.

For a generic study of evaporators a detailed comparison is made below.

8.1.1 Types of Evaporators:

Different types of evaporators are used in different types of refrigeration applications and accordingly they have different designs. Here we have classified the evaporators based on their construction.

8.1.2 Classification of the Evaporators Based on the Construction

1) Bare Tube Evaporators

The bare tube evaporators are usually used for liquid chilling. In the blast cooling and the freezing operations the atmospheric air flows over the bare tube evaporator and the chilled air leaving it used for the cooling purposes.

2) Plate Type of Evaporators

In the plate type of evaporators the coil usually made up of copper or aluminium is embedded in the plate so as so to form a flat looking surface. Externally the plate type of evaporator looks like a single plate, but inside it there are several turns of the metal tubing through which the refrigerant flows. The advantage of the plate type of evaporators is that they are more rigid as the external plate provides lots of safety.

3) Finned Evaporators

The finned evaporators are the bare tube type of evaporators covered with the fins. When the fluid (air or water) to be chilled flows over the bare tube evaporator lots of cooling effect from the refrigerant goes wasted since there is less surface for the transfer of heat from the fluid to the refrigerant. The fins are the external protrusions from the surface of the coil and they extend into the open space. They help removing the heat from the fluid that otherwise would not have come in contact with the coil.

8.2 EVAPORATOR DESIGN

8.2.1 Evaporator Design At new values of 0°C.

$Water T_i$ = 8°C

$Water T_o$ = 0°C

Cooling load = Q_{evap}=1408 kW

$\dot{m}_{ref} = 1.30\ kg/s$

8.2.2 Flow rate of water:

$Q = \dot{m}C_p\Delta T$

$\dot{m} = Q/C_p\Delta T$

=-1408/4.180(8-0)

=42.1 kg/s

8.2.3 Effectiveness (ε):

$C_h = m_h C_{ph}$

= 42.0*4.18

=175.6 kW/°K

$C_c = m_c C_{pc}$

=1.17*2.62

=3.06kW/°K

$C = \frac{C_{min}}{C_{max}}$

=3.06/175.6

=0.017

≈ 0

$Q_{max} = C_{min}(T_{hin} - T_{cin)}$ +(m*λ*$\varkappa$)

=3.06*(8-0) + (1.17 *1370*0.950)

=1523 kW

ε = $\frac{Q}{Q_{max}}$ = 1408/1523 = 0.924

8.2.4 NTU

Since C≈ 0

$$\varepsilon = 1 - e^{(-NTU)}$$

$NTU = -\ln(1-\varepsilon)$

$= -\ln(1-0.92)$

= 2.58

8.2.5 Ts (Surface Temperature):

$Q = \varepsilon m Cp(Ts - Ti)$

-1408 =0.924 * 42.1 *4.18 *(Ts - 281)

$$Ts - 281 = \frac{-1408}{0.924 * 42.1 * 4.18} 0$$

Ts=272 °K

=-0.659 °C

8.2.6 Overall Heat Transfer coefficient (U):

Assumptions:

- No fouling
- No ice formation on tubes

$h_{water} = 6809\ \frac{W}{m^2}$°K

$h_{Ammonia} = 1702\ \frac{W}{m^2}$°K

$k_{copper} = 399\ \frac{W}{m^2}$°K

$$U = \frac{1}{\frac{1}{h1} + \frac{\Delta\chi}{k} + \frac{1}{h2}}$$

$$U = \frac{1}{\frac{1}{1702} + \frac{3*10^{-3}}{399} + \frac{1}{6809}}$$

U=1308 W/m^2°K

$$U = \frac{1}{\frac{1}{ho} + \frac{do*ln(\frac{do}{di})}{2kw} + \frac{do}{hi*di}}$$

Assuming do= $50 * 10^{-3}$ m

di=47 $* 10^{-3}$ m

$$U = \frac{1}{\frac{1}{6809} + \frac{50*10^{-3} ln(\frac{50*10^{-3}}{47*10^{-3}})}{2*399} + \frac{50*10^{-3}}{1702*47*10^{-3}}}$$

U=1290 W/m^2°K

This means that our assumed values are close to actual values.

8.2.7 Surface Area:

$$As = \frac{NTU * Cmin}{U}$$

$$As = \frac{2.58 * 3.06 * 10^3}{1290}$$

As = 6.12 m^2

8.2.8 Total length of tube/coil:

$$As = \barwedge DL$$

$$L = \frac{As}{\barwedge D}$$

$$L = \frac{6.12}{3.14 * 50 * 10^{-3}}$$

L = 38.9 m

Now let n=52

$$As = n \barwedge DL$$

$$l = \frac{As}{n \barwedge D}$$

$$l = \frac{6.12}{52 * 3.14 * 50 * 10^{-3}}$$

$$l = 0.750m$$

So if 52 individual tubes or passes are to be used then length of each tube or pass should be 0.75

8.2.9 Pressure Drop:

$\Delta P = pi - po$

$$\Delta P = \frac{8fL\pi u^2}{2d}$$

We need to calculate Friction factor

$$N_{Re} = \frac{\pi D v}{\mu}$$

$$N_{Re} = \frac{3.08 * 47 * 10^{-3} * 0.600}{9.08 * 10^{-3}}$$

$N_{Re} = 9560$

Absolute roughness = 0.015* 10^{-3} (for copper)

From graph (moody diagram)

Relative roughness=$\frac{0.015*10^{-3}}{47*10^{-3}}$

Relative roughness = 3.19*10^{-5}

Now from graph and software

f=0.031

$$\Delta P = \frac{8fL\pi u^2}{2d}$$

$$\Delta P = \frac{8 * 0.031 * 38.9 * 3.08 * 0.60^2}{2 * 47 * 10^{-3}}$$

$\Delta P = 114\ Pa$

$\Delta P = 0.001\ bar$

8.3 CONDENSERS

In systems involving heat transfer, a condenser is a device or unit used to condense a substance from its gaseous to its liquid state, by cooling it. In so doing, the latent heat is given up by the substance and transferred to the surrounding environment. Condensers can be made according to numerous designs, and come in many sizes ranging from rather small (hand-held) to very large (industrial-scale units used in plant processes).

There are several types

1. **Air Cooled**

If the condenser is located on the outside of the unit, the air cooled condenser can provide the easiest arrangement. These types of condensers eject heat to the outdoors and are simple to install.

2. **Water Cooled**

Although a little more pricey to install, these condensers are the more efficient type. Commonly used for swimming pools and condensers piped for city water flow, these condensers require regular service and maintenance.

They also require a cooling tower to conserve water. To prevent corrosion and the forming of algae, water cooled condensers require a constant supply of makeup water along with water treatment.

3. **Evaporative Condenser**

While these remain the least popular choice, they are used when either water supply is inadequate to operate water cooled condenser or condensation temperature is lower that can achieved by air cooled condenser. Evaporative condensers can be used inside or outside of a building and under typical conditions, operate at a low condensing temperature.

Engro uses the evaporator condenser which is explained in detail below.

8.3.1 Principle of Operation

The vapour to be condensed is circulated through condensing coils, which is continually wetted on the outside by water sprayed over tubes- Air is pulled over the coil, causing a small portion of the water to evaporate. The evaporation removes latent heat from the vapour in the coil, causing it to condense.

8.3.2 Types of Evaporator condensers

1) According to directions of Air and “later Flow

- Combined Flow: Combined flow is the use of both a condensing coil and fill surface for heat transfer in an evaporative condenser. The addition of fill surface to the traditional evaporative condenser design reduces evaporation in the coil section reducing the potential for scaling and fouling- Combined flow evaporative condensers utilize parallel flow of air and spray water over the coil, and cross flow air/water flow through the fill surface.

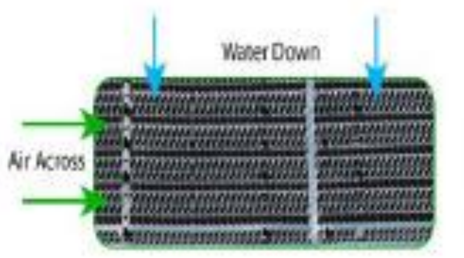


Figure 2 Combined flow of evaporator condensers

Parallel Flow: In parallel flow, air and water flow over the coil in the Same direction In the fill section of BAC's combined flow evaporative condensers. Air and water interact in a cross flow

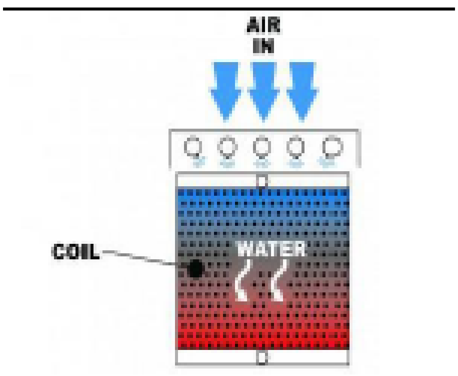


Figure 3 Parallel flow of evaporator condensers

- Counter Flow: In counter flow evaporative condenser design. The flow of the Air in the opposite direction of the spray water. In counter flow evaporative condensers. air travels vertically up through the unit while the spray water travels vertically down over the coil-

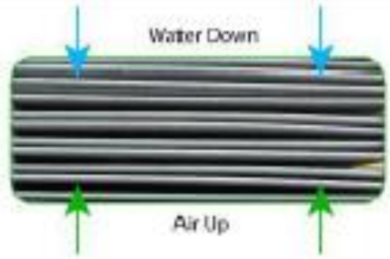


Figure 4 Counter flow of evaporator condensers

2) According to the type of Fan used

The flow of air through most factory assembled evaporative condensers is provided by one or more mechanically driven fans. The fans may be axial or centrifugal. Each type having its own distinct advantages.

- Axial Fan: Axial fan units require approximately half the fan motor horsepower comparably sized centrifugal fan units, offering significant life-cycle cost savings.

Figure 5 Axial Fan

- Centrifugal Fan : Centrifugal fan units are capable of overcoming reasonable amounts of external static pressure making them suitable for both indoor and outdoor installations. Centrifugal fans are also inherently quieter than axial fans. although the difference is minimal and can often be overcome through the application of optional low sound fans and or sound attenuation on axial fan units.

CG fans are capable of overcoming reasonable amounts of external static pressure making them suitable for both indoor and outdoor installations. CG fans are also inherently quieter than axial fans, although the difference is minimal.

Figure 6 Centrifugal Fan

3) According To Draft

- Forced Draft: Rotating air handling components are located on the air inlet face at the base of forced draft units, facilitating easy access for routine maintenance and service. Additionally, location of these components in the dry entering air stream
- Induced Draft: The rotating air handling components of induced draft equipment are mounted in the top deck of the unit, minimizing the impact of fan noise on near-by neighbors and providing maximum protection from fan icing with units operating in sub-freezing conditions- The use of corrosion resistant materials ensures long lifeand minimizes maintenance requirements for the air handling components.

8.4 CONDENSER DESIGN

8.4.1 GIVEN VALUES OF NH_3 at 97.3°C

ρ = 8.29 kg/m^3

H = 1640kJ/kg

$C_p = 2.59\frac{kJ}{kgK}$

$\mu = 12.7\mu Pa.s$

K = 0.0345 W/m°K

Pr= 0.957

S= 5.43 $\frac{kJ}{kgK}$

8.4.2 GIVEN VALUES OF NH_3 at 34°C

ρ = 589 kg/m^3

H = 342 kJ/kg

$C_p = 4.87 \frac{kJ}{kgK}$

$\mu = 121\ \mu Pa.s$

K = 0.46 W/m°K

Pr= 1.28

8.4.3 GIVEN VALUES OF Air at 35°C

Wet Bulb T = 28°C

Relative H = 60%

Specific Humidity = 0.0211

Specific Volume = 0.9016 $\frac{m^3}{kg}$

H = 89.33 kJ/kg

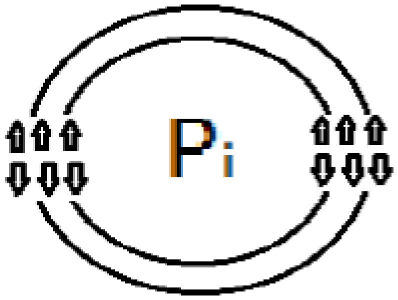
8.4.4 Water Values

Inlet T = 30°C

Outlet T= 33.5°C

H_{in}= 125.7 kJ/kg

H_{out} = 140 kJ/kg



According to thin cylindrical formula

$P_i * D * L = \tau * t * 2 * L$

$P_i * D = 2 * t * \tau$

Where $P_i = internalPressureof\ 14\ bar$

$\tau = tensile\ stress\ of\ SS316 = 515MPa$

$D_o = 26mm$ (Assumed)

t =$(P_i * D)/(2 * \tau)$

t = 1.54 mm

8.4.5 FINAL TUBE DIMENSIONS

$D_o = 26mm$ (Assumed) and $D_i = 24.5$

$t = 1.54\ mm$

2. Velocity of Ammonia

At 97.3°C

$$v = \frac{\dot{q}}{A_i}$$

$$v = \frac{\dot{m}}{\rho * A_i}$$

Where the values of $\rho$and A_i are given.

$$v = 340\ \frac{m}{s}$$

At 34°C

$$v = \frac{\dot{q}}{A_i}$$

$$v = \frac{\dot{m}}{\rho * A_i}$$

$$v = 4.87 \frac{m}{s}$$

8.4.6 Heat Transfer Coefficient for inside of the tubes

Find $\boldsymbol{N_{Re}}$ for inlet ammonia (97.3°C)

$$N_{Re} = \frac{\rho D v}{\mu}$$

$$N_{Re} = \frac{(8.29) * (346) * (0.024)}{12.7 * 10^6}$$

$$N_{Re} = 5.53 * 10^6$$

Find $\boldsymbol{N_{Re}}$ for outlet ammonia

$$N_{Re} = \frac{\rho D v}{\mu}$$

$$N_{Re} = \frac{(589) * (4.87) * (0.024)}{121 * 10^6}$$

$$N_{Re} = 0.580 * 10^6$$

Avg $N_{Re} = 3.056 * 10^6$

For smooth tubes 'f' in turbulent flow is found using the Equation:

$$f = (0.79 * \ln(N_{Re}) - 1.64)^{-2}$$

$$f = 9.69 * 10^{-3}$$

$$Nu = 0.125 * f * N_{Re} * (Pr)^{\frac{1}{3}}$$

$$Nu = 3950$$

$$\mathrm{Nu} = \frac{\mathrm{hD}}{\mathrm{k}}$$

$$Nu = \frac{h_i D}{k}$$

$$h_i = \frac{Nu * k}{D}$$

h_i = 37500W/m^2°K

8.4.7 Heat Transfer Coefficient for inside of tubes

Lest assume the tube arrangement

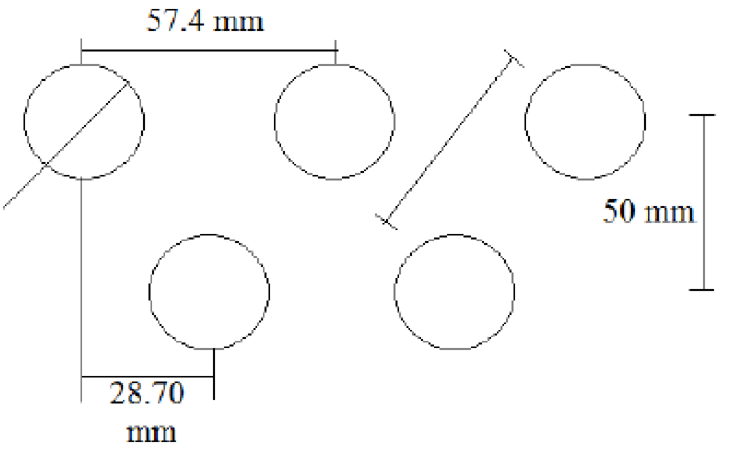


Figure 7 Tube arrangement of Condenser

With

D_o= 0.026m

S_T =57.4 mm

$S_L = 50\ mm$

Lets also assume length of condenser tubes = 6m

Tube bundle width = 3m

Tube pitch = 57.4 mm

So 1 layer of tube bundle has = 3000/ 57.4

= 52 tubes

Area of 1 tube = $2 * \pi * D * L$

Area = 0.98 m^2

To find H_o *:*

$H_o = 2102.9 * (\frac{\gamma}{d_t})^{1/3}$

$$\frac{\gamma}{d_t} = \frac{m * d_t}{2 * n * p * l}$$

Where n=300 tubes, l = 6m, p = 57.44 mm, $d_t = 1$

$$\frac{\gamma}{d_t} = 0.595$$

$H_o = 1770$ W/m^2°K

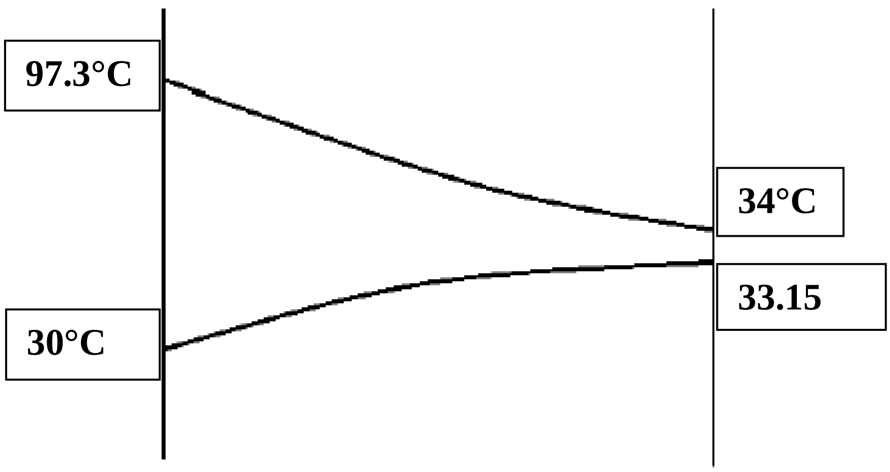
8.4.8 Overall Heat Transfer Coefficient (U)

$U * A = \frac{1}{\frac{1}{h_i \pi D} + \frac{ln\,(\frac{D_o}{D_i})}{2\pi k L} + \frac{1}{h_o \pi D}}$

Substituting all the values

$U = 1440$ W/m^2°K

8.4.9 Calculating the Area of De-superheating and Phase Change



$\Delta H = 1640 - 342 = 1298 \frac{kJ}{kg}$

$$LMTD = \frac{(t_1 - t_2) + (T_2 - T_1)}{\ln\left(\frac{t_1 - T_1}{t_2 - T_2}\right)}$$

Where

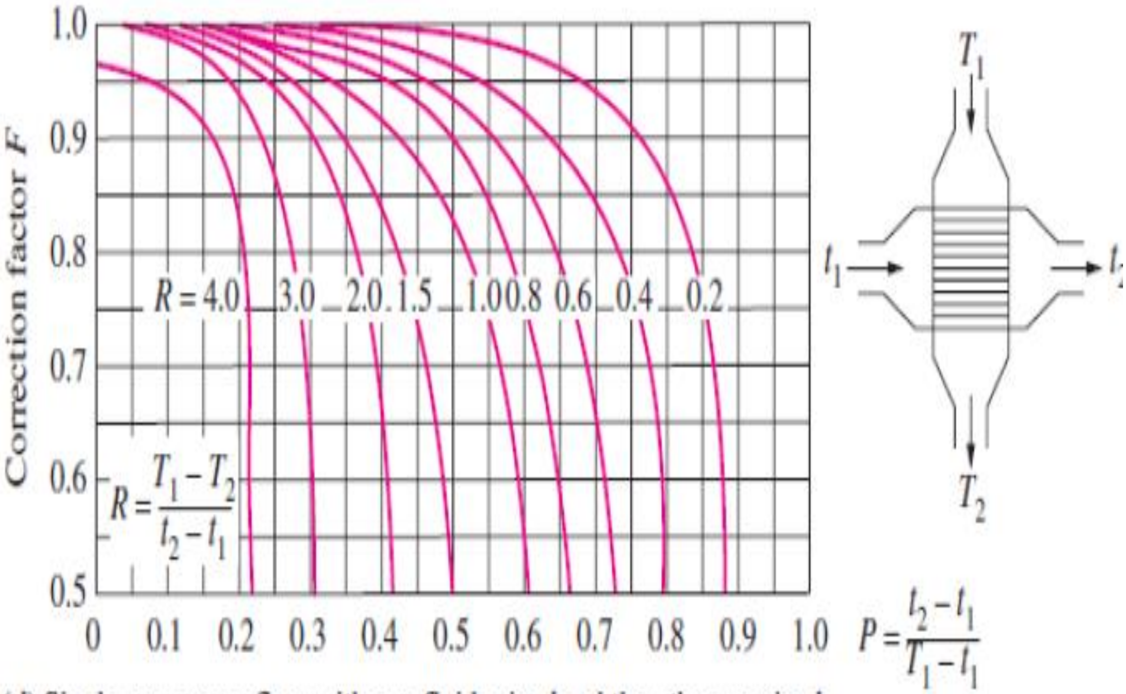
$t_1 = 97.3°C$

$t_2 = 34°C$

$T_1 = 30°C$

$T_2 = 33.15°C$

$LMTD = 15.2°C$



(*d*) Single-pass cross-flow with one fluid *mixed* and the other *unmixed*

Using these correlations

R= 0.0498

P = 0.94

From the graph

f = 0.98

So $LMTD_f$ = 14.896

Using

$Q = U * A * LMTD$

A = 14.8 m^2

For Phase Change Section Use AMTD

$$AMTD = \frac{(t_1 - T_1) + (T_2 - T_1)}{2}$$

$AMTD = 3.575$

Using the same equation

A = 323.6 m^2

Total Area = 338.4 m^2

So our assumptions of 300 tubes in nearly correct.

Chapter 9 COMPUTATIONAL ANALYSIS

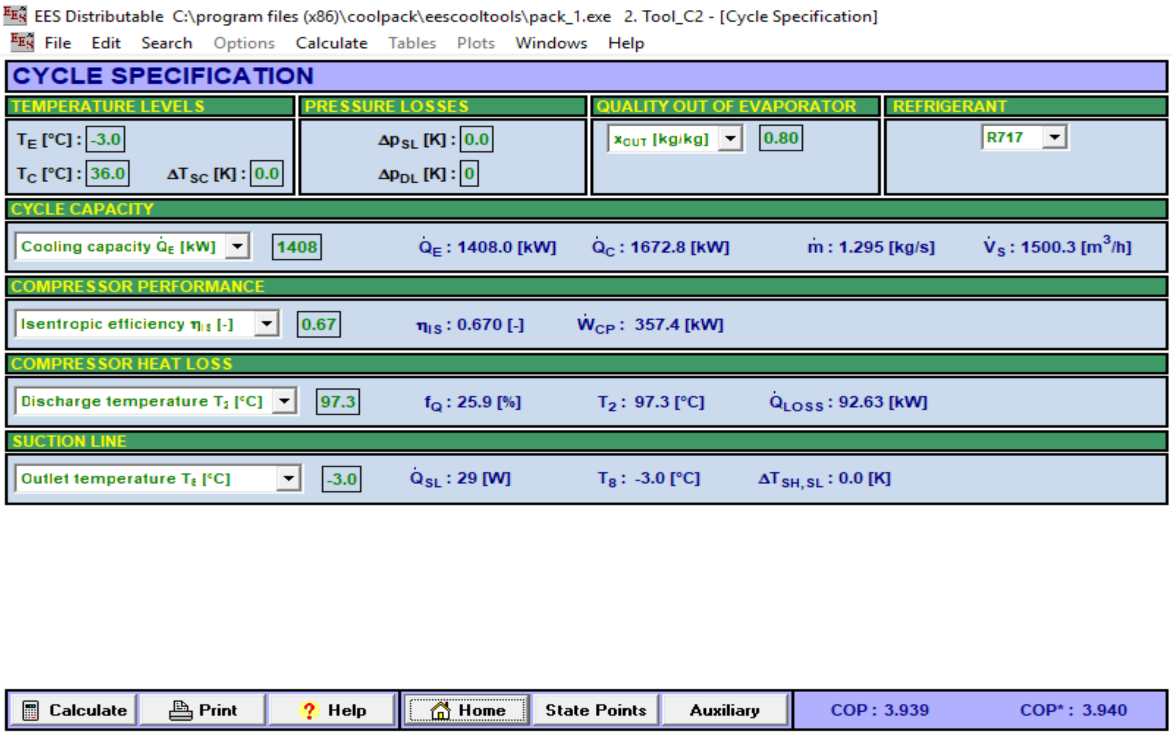
9.1 SOFTWARES USED; COOLPACK

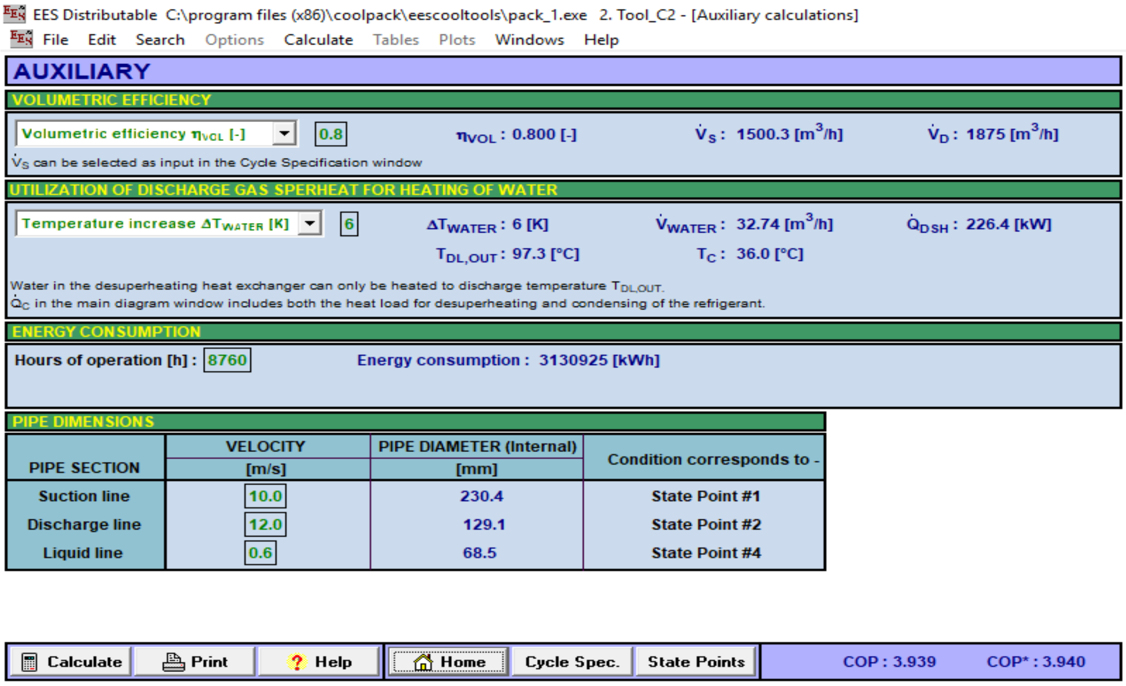
CoolPack is used to model the pH diagram using any of the following refrigerants: R717, R134a, etc. Given the working conditions, CoolPack models the diagram giving an idea of the energy that is utilized in the refrigeration cycle, the suction and discharge line diameter, and the liquid line diameter.

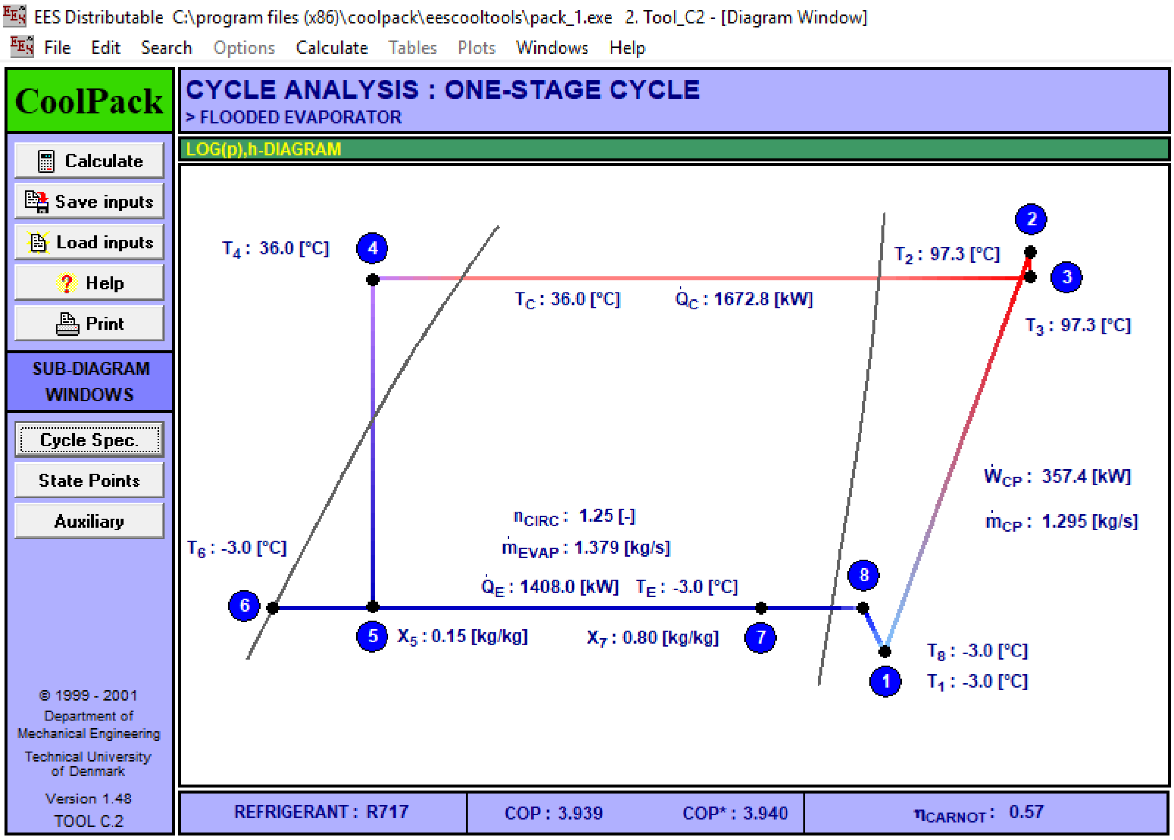
Using this software has enabled us to model the pH diagrams for various refrigerants and conduct a comparative study based on the energy utilized and the COP of the system. This is a huge advantage when it comes to choosing refrigerants when they are of similar price.

9.2 RESULTS

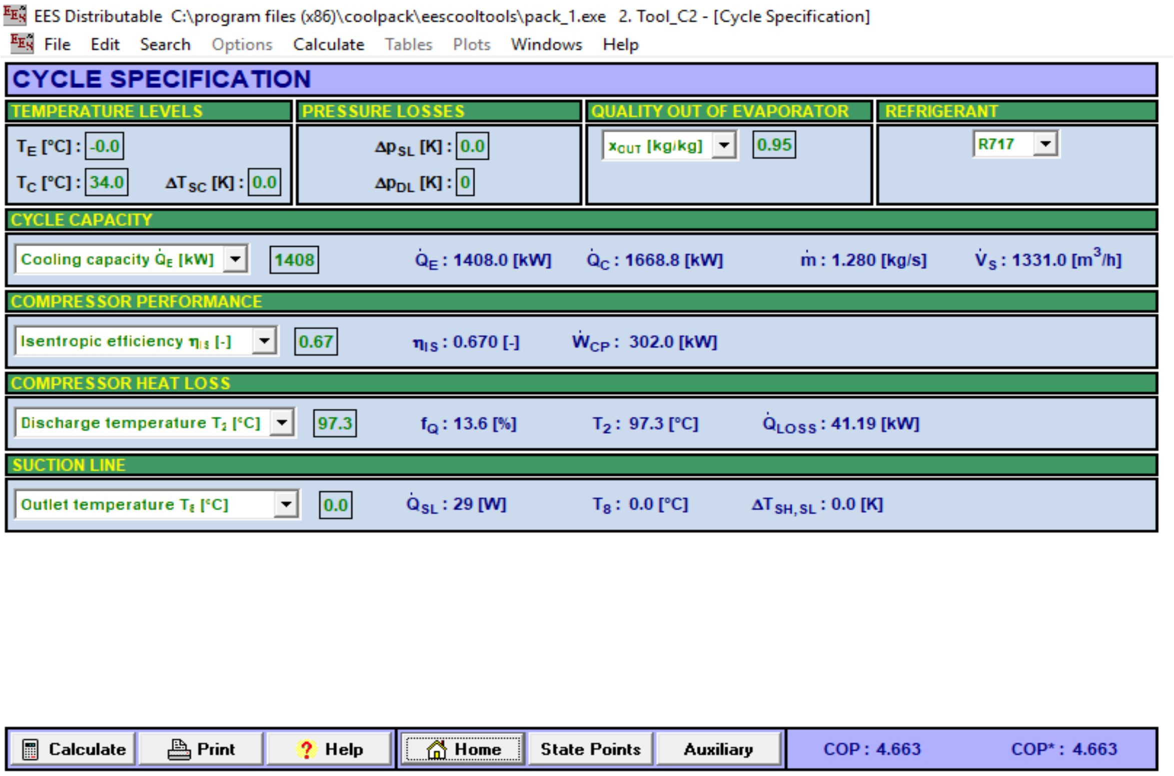
Based on working conditions:

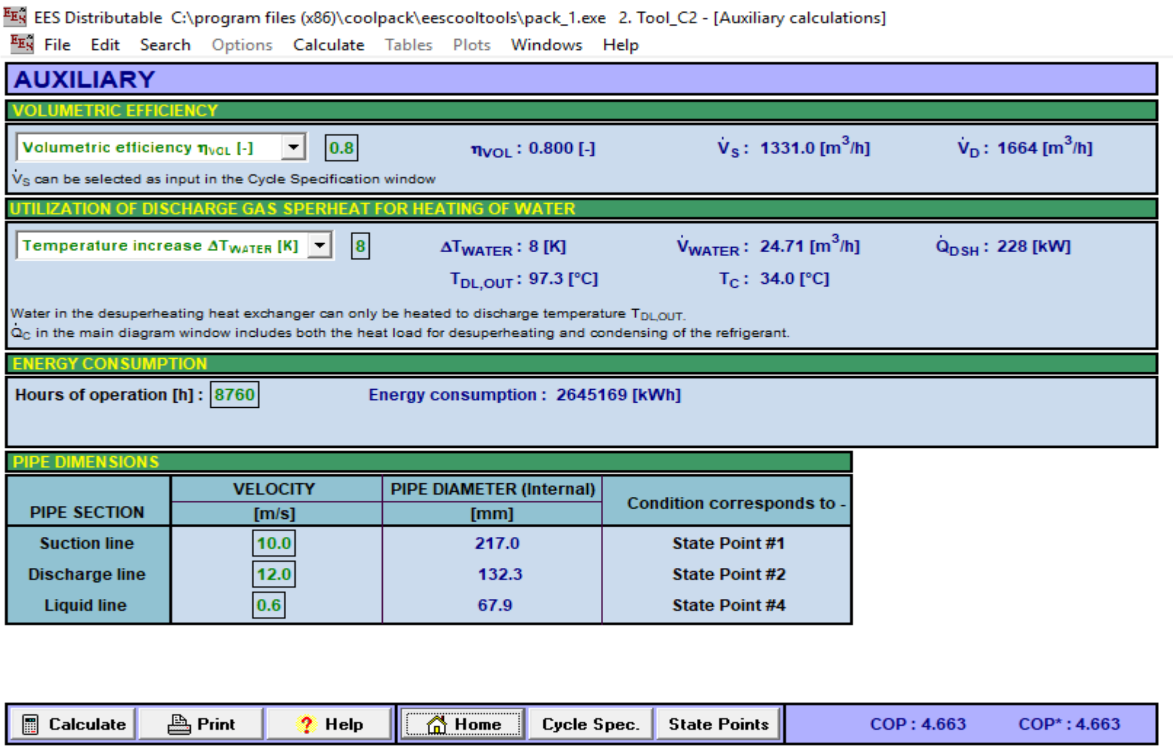


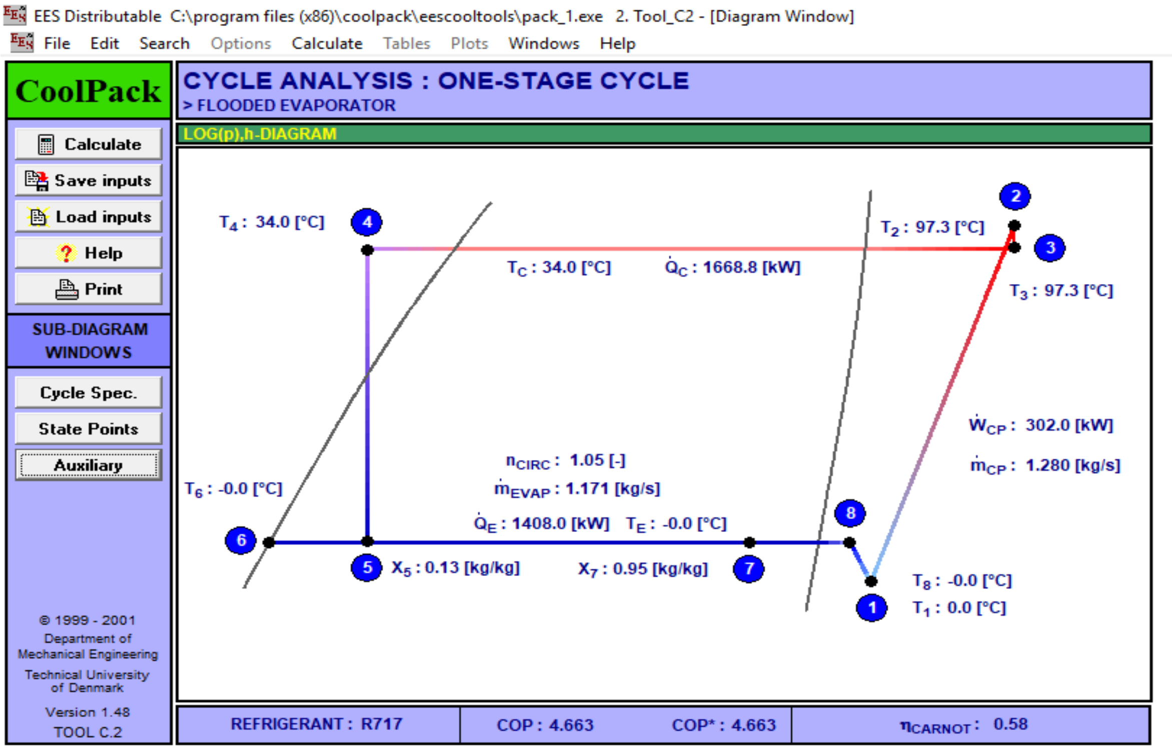

EES Distributable C:\program files (x86)\coolpack\eescooltools\pack_1.exe 2. Tool_C2 - [Auxiliary calculations]
File Edit Search Options Calculate Tables Plots Windows Help
AUXILIARY
VOLUMETRIC EFFICIENCY
Volumetric efficiency η_{VOL} [-] 0.8
η_{VOL} : 0.800 [-]
$\dot{V}_S$: 1500.3 [m³/h]
$\dot{V}_D$: 1875 [m³/h]
$\dot{V}_S$ can be selected as input in the Cycle Specification window
UTILIZATION OF DISCHARGE GAS SPERHEAT FOR HEATING OF WATER
Temperature increase ΔT_{WATER} [K] 6
ΔT_{WATER} : 6 [K]
$\dot{V}_{WATER}$: 32.74 [m³/h]
$\dot{Q}_{DSH}$: 226.4 [kW]
$T_{DL,OUT}$: 97.3 [°C]
T_C : 36.0 [°C]
Water in the desuperheating heat exchanger can only be heated to discharge temperature $T_{DL,OUT}$.
$\dot{Q}_C$ in the main diagram window includes both the heat load for desuperheating and condensing of the refrigerant.
ENERGY CONSUMPTION
Hours of operation [h] : 8760
Energy consumption : 3130925 [kWh]
PIPE DIMENSIONS
PIPE SECTION
VELOCITY [m/s]
PIPE DIAMETER (Internal) [mm]
Condition corresponds to -
Suction line 10.0 230.4 State Point #1
Discharge line 12.0 129.1 State Point #2
Liquid line 0.6 68.5 State Point #4
Calculate
Print
Help
Home
Cycle Spec.
State Points
COP : 3.939
COP* : 3.940


EES Distributable C:\program files (x86)\coolpack\eescooltools\pack_1.exe 2. Tool_C2 - [Diagram Window]
File Edit Search Options Calculate Tables Plots Windows Help
CoolPack
CYCLE ANALYSIS : ONE-STAGE CYCLE
> FLOODED EVAPORATOR
LOG(p),h-DIAGRAM
Calculate
Save inputs
Load inputs
Help
Print
SUB-DIAGRAM WINDOWS
Cycle Spec.
State Points
Auxiliary
© 1999 - 2001
Department of Mechanical Engineering
Technical University of Denmark
Version 1.48
TOOL C.2
T_4 : 36.0 [°C]
T_2 : 97.3 [°C]
T_C : 36.0 [°C]
$\dot{Q}_C$: 1672.8 [kW]
T_3 : 97.3 [°C]
$\dot{W}_{CP}$: 357.4 [kW]
n_{CIRC} : 1.25 [-]
$\dot{m}_{CP}$: 1.295 [kg/s]
T_6 : -3.0 [°C]
$\dot{m}_{EVAP}$: 1.379 [kg/s]
$\dot{Q}_E$: 1408.0 [kW]
T_E : -3.0 [°C]
X_5 : 0.15 [kg/kg]
X_7 : 0.80 [kg/kg]
T_8 : -3.0 [°C]
T_1 : -3.0 [°C]
REFRIGERANT : R717
COP : 3.939
COP* : 3.940
η_{CARNOT} : 0.57

9.3 Based on new design:







9.4 COMPARATIVE STUDY OF REFRIGERATNTS:

There are no refrigerants in the horizon that completely meet the safety, stability, energy efficiency and environmental friendliness. It seems that the refrigeration industry will have very little choice but to use flammable refrigerants (HFOs. Low GWP HFCs. HCs. NH3. Etc). Since the energy efficiency of HFOs is somewhat low, mixtures of medium GWP fluids such as R32 and low GWP refrigerants such as R1234yf may be the working fluids of choice in the immediate future. Meanwhile the quest for better molecules continues. Barring new inventions, natural refrigerants appear to be the best choice in the long term.

The following table enlists the different refrigerants and their thermodynamic properties. One very important factor to consider is the boiling point and for a good refrigerant it has to be as low as to be compatible with both the Evaporator and Condenser. Higher boiling points will not vaporise easily and so cannot be efficiently used.

Other factors are the environmental considerations. 2 important factors are the **ODP-ozone depletion potential** and the **GWP-Global warming potential.** Both have to be as low as possible for minimum impact on the environment.

No.	Refrigerant	Boiling Point @1 atm(K)	Freezing Point (K)	Critical Temp (K)	Critical Pr (bar)	ODP	GWP
CFCs (Chlorofluorocarbons)							
R113	Trichlorotrifluroethane	320.73	238.16	487.3	34.4	0.9	5200
R11	Trichloroflurometh ane	296.98	162.05	471.2	44.1	1	4000
R114	Dichlorotetrafluroethane	276.94	179.27	418.9	32.6	0.7	16600
R12	Dichlorodiflurometh ane	243.37	115.38	385.2	41.2	1	12200
R115	Chloropentafluroethane	233.83	167.05	353.1	31.5	0.6	39200
HCFCs(Hydrochlorofluorocarbons)							
R141b	Dichlorofluroethane	305.16	–	483.35	46.4	0.15	600
R123	Dichlrotrifluroethane	301.03	166.01	457.15	36.76	0.02	80
R22	Chlorodiflurometh ane	232.40	113.16	363.15	49.78	0.05	1480
HFCs (Hydrofluorocarbons)							
R245fa	Pentafluropropane	288.44	166.49	383.4	31.5	0	790
R134a	Tetrafluroethane	247.00	176.55	374.25	40.67	0	1160
R507	Azeotrope - Blend	226.05	255.38	344.05	37.92	0	1400
R125	Pentafluroethane	224.59	170.01	339.25	36.2	0	3360
R32	Diflurometh ane	221.44	137.05	351.4	58.08	0	440
R23	Triflurometh ane	191.10	118.16	298.75	48.37	0	24000
HFOs (Hydrofluorooellifins)							
R1234yf	2,3,3,3-Tetrafluoropropene	244.15	220.00	367.85	33.82	0	4
FCs /PFCs (Fluorocarbons/Perfluorocarbons)							
R218	Octofluropropane	241.66	113.16	344.95	26.8	0	9300
R14	Tetraflurometh ane	145.22	89.27	227.65	37.43	0	6500

Hydrocarbons							
R600	Butane	272.66	134.66	425.12	37.7	0	0
R290	Propane	231.07	85.49	369.83	42.1	0	0
R170	Ethane	184.35	90.38	305.32	48.5	0	0
R1150	Ethylene	169.44	104.27	282.34	50.3	0	0
R50	Methane	111.66	90.94	190.56	45.9	0	0
Inorganic Compounds							
R718	Water	373.16	273.16	647.13	219.4	0	0
R717	Ammonia	239.83	195.44	405.65	113.0	0	0
R744	Carbon dioxide	194.72	216.55	304.21	73.9	0	1
R728	Nitrogen	77.38	63.16	126.2	33.9	0	0
R702n	Hydrogen	20.38	13.99	33.19	13.2	0	0
R704	Helium	4.22	_	5.2	2.3	0	0
HFEs (Hydrofluoroethers)							
HFE-7100	Methoxynonafluorobutane	334.16	138.16	468.45	22.3	0	320
HFE-7200	Ethoxynonafluorobutane	349.16	135.16	482.0	19.8	0	55
HFE-7000	Methooxyheptafluropropane	307.16	150.38	438.15	24.8	0	400

9.4.1 Different Refrigerants

1. CFCs
2. HFC
3. HFO (hydrofluoroolefins)
4. FIC (fuoroiodocarbons
5. Hydrocarbons
6. Ammonia
7. Water
8. Air
9. Carbon-Dioxide

In the meantime there are some easy options to move to lower GWP refrigerants now. R404A is one 01 the most commonly used refrigerants in Europe yet it has a high GWP (3922). Products such as Performax LT (R407F) not only have much lower GWP (1824) but also offer worthwhile energy savings making it of interest for new and existing equipment instead of R404A.

CFCs pose a great threat to the ozone because of the presence of Chlorine in them. A major shift to HFCs can prove both an effective approach and a new area of study in research. The Natural refrigerants like Air, Water and Carbon dioxide are also looked in but these do not provide a higher refrigeration capacity than the other refrigerants.

Despite extensive research and development there is no single refrigerant that satisfies all these criteria and selection involves some level of compromise.

The following figure lists the different generations of Refrigerant from the first generation to the fourth one.

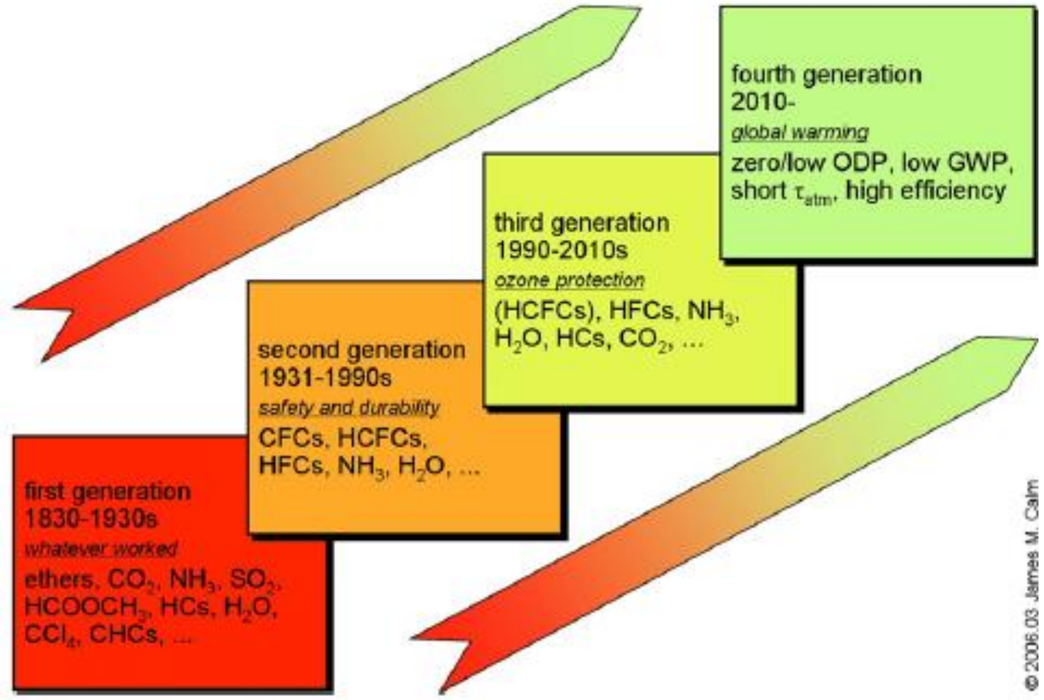
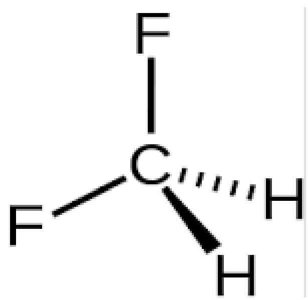


Figure 8 Different Generations Of Refrigerant

9.4.2 R-32 – A potential Refrigerant

R32 is Difluoromethane with the following structure



It is a chlorine-free, ozone-safe fluorocarbon whose boiling point is -52°C. R32 is a component of R410A, a blend refrigerant widely used in developed countries as a main alternative to HCFC-22, an ozone depleting refrigerant. But R410A has a high Global Warming Potential (hereafter, GWP) of 2088, so a new refrigerant with lower GWP is needed to mitigate climate change. R32 has a GWP about one third that of R 410A, and it has excellent properties as a refrigerant. Therefore, the technology has been developed to use it by itself as an alternative refrigerant to replace R410A.

Product		R32	R410A	HCFC-22
Component		HFC-32	HFC-32/ HFC-125	HCFC-22
Chemical formula		CH_2F_2	CH_2F_2/ CHF_2CF_3	$CHClF_2$
Composition	mass%	100	50/50	100
Molar mass		52.0	72.6	86.5
Boiling point		-51.7	-51.4	-40.8
Freezing point		-136	-	-160
Critical temperature		78.1	72.0	96.2
Critical pressure	MPa	5.78	4.95	4.99
Critical density	kg/m^3	424	486	515
Density Saturated liquid	kg/m^3	961	1059	1191
Density Saturated vapor	kg/m^3	47.34	64.87	44.23
Viscosity Saturated liquid	mPa·s	0.116	0.121	0.178
Viscosity Normal pressure vapor	mPa·s	0.0126	0.0129	0.0128
Isobaric specific heat Saturated liquid	kJ/kg·K	1.937	1.711	1.256
Isobaric specific heat Normal pressure vapor	kJ/kg·K	0.848	0.818	0.662
Latent heat of vaporization (Boiling Point)	kJ/kg	382	275	233
Thermal conductivity Saturated liquid	mW/m·K	125	87	87
Thermal conductivity Normal pressure vapor	mW/m·K	13	13	11
Breakdown voltage Normal pressure vapor	kV	2.8	4.8	7.2
Dielectric constant Saturated liquid		14.27	7.88	6.35
Acceptable concentration limit	ppm	1000[*2]	1000[*3]	1000[*4]
Ozone depletion potential ODP	CFC11=1	0	0	0.055
Global warming potential GWP [*1]	CO_2=1	675	2088	1810
Solubility of water	massppm	3400	1600	1300

Flammability

R32 is flammable, but its flammability is extremely low compared with that of hydrocarbon refrigerants such as propane. Therefore, R32 is positioned as a slightly flammable refrigerant.

Saturated Vapor Pressure / Temperature Curves

The graph below shows the relationship between temperature and saturated vapour pressure of R32, in comparison with that of R410A and HCFC-22. As shown on the graph, R32 has similar vapor pressure to that of R410A.

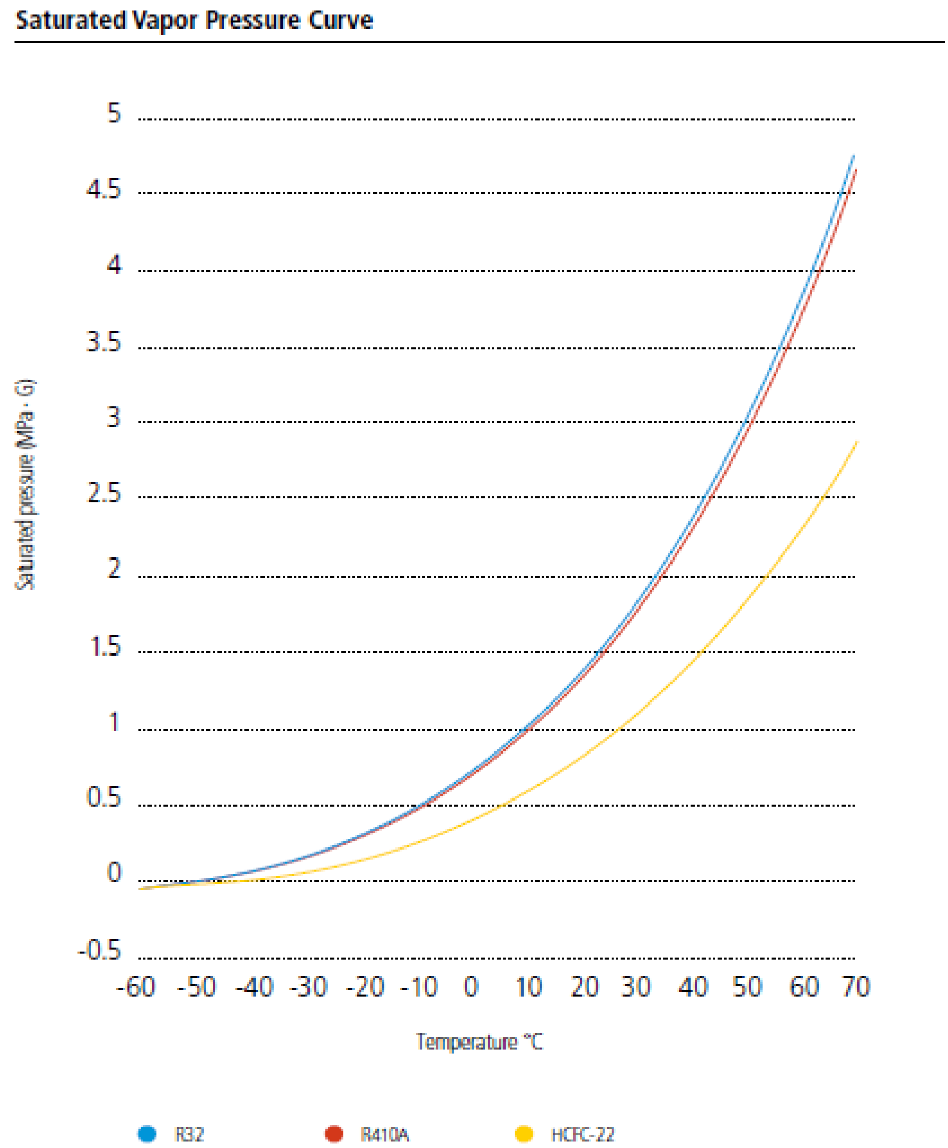


Figure 9 Relationship between temperature and saturated vapour pressure of R32

Theoretical characteristics of the refrigeration cycle

R32 delivers superior performance in both cooling / heating capacity and energy efficiency – compared with R410A, volumetric capacity of R32 is about 15% higher and its COP is about 6% higher (therefore, concerning the climate change issue, it can contribute to reduce the equipment's indirect impact on CO2 emission). But the discharge gas temperature of R32 is about 20°C higher, so this feature must be taken into consideration in equipment design.

<u>Price</u>

R32 has a higher price than ammonia which will increase the plant's initial cost but since it uses smaller compressors because of its higher density than ammonia the operating cost of the plant decreases. Also the compressor capital cost will also decrease. R-32 provides a higher refrigeration capacity than R-717 with the suction side of a refrigerating system at positive pressure down to -50°C.

<u>Conclusion</u>

With the above considerations, R32 is being used in the refrigeration industry other than ammonia. At Engro's plant site, R32 can be used but this has a long way to go. There would be a complete change in the design specifications of all the equipments being used and the budget of the plant.

It offers the prospect of miscible lubricants used with semi-hermetic motor-compressors and could lead to simpler, cheaper industrial installations. However the refrigerant itself is not as cheap as R-717 and it could still become a target of climate change phase down limits as the GWP is 675, classed as "moderate GWP" in the UNEP RTOC report (2014).

The prospect of significant displacement of R-32 by R-717 in the industrial market seems however remote in the short term future because

- Hazard assessment requirements are less well understood
- There is a fear of future HFC phase down
- Industrial equipment is not widely available
- End users are not unwilling to use R-717 in industrial systems
- R-717 has a good track record of safety, efficiency, ease of maintenance and reliability

A significant shift towards greater use of R-32 in the industrial market would therefore only be likely to happen if there was a strong imperative to move away from R-717. This would most likely be on grounds of toxicity but since ammonia is so widely used in other industries and is so familiar and well-understood a sudden move against it seems improbable.

9.5 Using Variable Frequency Drives (VFDs)

Refrigeration systems are designed for full-load conditions. Most of the time, however, their loads are average, not peak, and full motor capacity is not required. During average conditions, motors in traditionally designed systems (without VFDs) are constantly running at a higher speed than necessary or frequently cycling on and off. Producing more capacity than needed wastes considerable energy, and frequent on/off cycling accelerates wear and shortens the useful life of motors, contactors, and other components. Frequently starting and stopping motors and continually accelerating them to full speed eliminate opportunities for reducing energy costs. VFDs can help in both of these areas as well as provide better product environments.

Although maximum-load conditions — high ambient temperature, high humidity, and fully loaded store fixtures and storage boxes — exist as little as 4 percent of running time, refrigeration systems must be designed to cope with them. However, it is just as desirable for systems to provide as much of the required capacity for expected part-load conditions as possible. Unfortunately, providing capacity to meet peak demands wastes considerable energy when part-load conditions exist, if machines simply are switched on and off.

A VFD is an electronic controller device capable of adjusting speed of electric motor by modulating power being delivered. It can

- Vary motor speed to match variable load requirement
- Soft start to eliminate mechanical, electrical, hydraulic surges/transients
- Potentially save energy, energy cost

There are 2 basic types

1. Variable torque – required torque increases as speed increases (condenser, evaporator fans)
2. Constant torque – required torque independent of speed (compressors)

Single speed fan with on/off control

- Most common method of head pressure control
- Need to set cut-in (e.g. 150 psig), cut-out pressures (e.g. 145 psig)

- Simple control method except:
    - Highest Energy Consumption
    - High maintenance (Fan Motors, belts)
    - Potential for liquid management problems in multiple condenser systems

VFDs can

- Set target head pressure,
- Modulate fan speed to maintain
- Simple to implement
- Slightly higher capital cost versus fixed speed
- Lowest energy consumption control alternative
- Multiple condenser systems, modulate ALL condensers together
- Smooth system operation with minimal transients

COMPARATIVE CONDENSER FAN PERFORMANCE

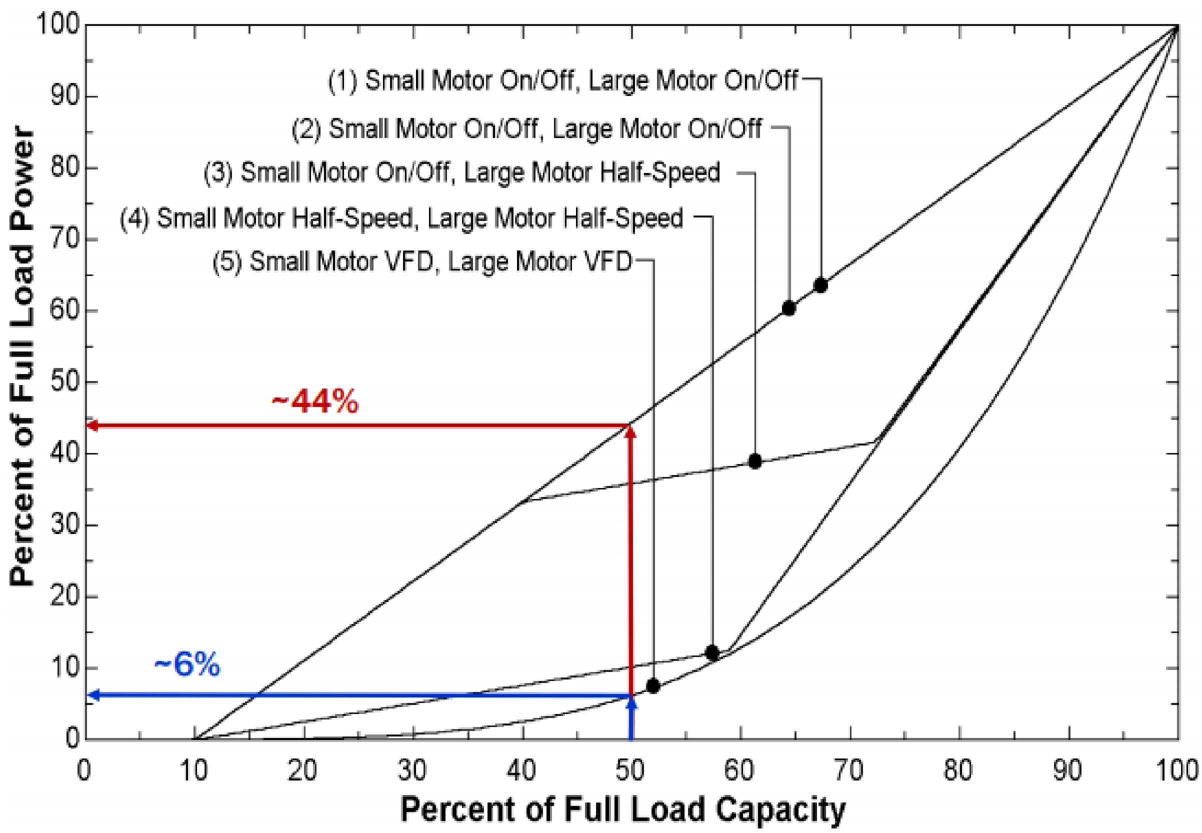


Figure 10 Comparative Condenser Fan Performance

The benefits of VFDs in Condenser

- Stable head pressure
- Reduced maintenance on shafts, bearings, belts
- Allows ability to optimize head pressure
- Reduced energy consumption and operating costs

VFDs in Compressors

The following essential advantages are obtained by continually adapting the power of a compressor pack by controlling the speed of a variable speed compressor: Improved cooling quality by maintaining a constant suction pressure

- Wider range of operation of the refrigeration power
- Increased power by increasing the speed of the variable speed compressor
- Energy saving
- Longer compressor lifetime
- Better possibilities of providing monitoring, remote setting and diagnostics

Reaching a high range of variation between maximum and minimum frequency is essential for optimum control performance with minimum deviation from the pressure set point. This can be made clear when considering the so-called Control Factor CF, which is:

Control Factor CF = Variation of VsC refrigeration power / Fixed-speed Compressor

Assuming that the variable speed compressor and fixed speed compressor piston displacement is equal and no cylinder off-loading is activated, this formula can be reduced to

Control Factor CF = (fmax [Hz] – fmin [Hz]) / 50 Hz

Where;

FsC: Fixed-speed Compressor.

VsC: Variable-speed Compressor.

A control factor of > 80 % is to be aimed for. A control factor less than 80 % results in an unstable suction pressure with poor control performance of the expansion valves.

There are other good reasons for increasing the maximum frequency as high as possible:

- Reduced number of starts of the fixed speed compressor (FsC). Every start is a strain for the refrigeration compressor bearings and motor windings.
- Compared to conventional rack installations without variable frequency drives, the reduced number of starts extends compressor lifetime considerably.
- The refrigeration capacity available is approximately proportional to the frequency. Operation at higher frequencies means that a higher refrigeration capacity can be achieved with the same installation.

CONCLUSION

Variable frequency drive technology is an essential component of state-of-the-art refrigeration technology. Users who are both open and interested will soon get up to speed regarding the requirements placed on the system planning and implementation.

Chapter 10 HAZOP ANALYSIS OF AMMONIA REFRIGERATION SYSTEMS

10.1 BACKGROUND

Ammonia (NH3) has been used as a refrigerant since the nineteenth century. Today food processing and cold storage industries are the main users of ammonia refrigeration systems. New applications using ammonia as refrigerant are under development and their use is expected to increase because of the thermodynamic and environmental characteristics of ammonia.

The advantages of ammonia as refrigerant include: low molecular weight (17.03). low boiling point (-28 “F at 0 psig) and high latent heat of vaporization (1371.2 kJ/kg at boiling point and 1.013 bar). Also ammonia has environmental advantages because it is not considered a greenhouse gas and it has an Ozone Depletion Potential (GDP) of 0.00 when released to the atmosphere [1]. These characteristics make ammonia an efficient and environmentally friendly refrigerant. In contrast. Fluorocarbon based refrigerants are under severe environmental regulations and the costs of installation and operation are higher than those for ammonia refrigeration systems.

However ammonia is toxic, flammable, explosive and corrosive. Table 1.1 presents a summary of the properties of ammonia. Several incidents have occurred in ammonia refrigeration facilities but well designed and maintained facilities have good safety records [3]. OSHA’s Process Safety Management program (PSM) and EPA's Risk Management Program (RMP) are mandatory for large facilities using ammonia as refrigerant [2]. Nevertheless. risk assessment is required regardless the size of the ammonia refrigeration system.

10.2 PROPERTIES OF AMMONIA

Boiling Point	-28 F
Weight per gallon of liquid at -28 F	5.69 pounds
Weight per gallon of liquid at 60 F	5.15 pounds
Specific gravity of the liquid (water=1)	0.619
Specific gravity of the gas (air=1)	0.588
Flammable limits in air	16-25%
Ignition temperature	1204 F
Vapor pressure at 0 F	16 psi
Vapor pressure at 68 F	110 psi
Vapor pressure at 100 F	198 psi
One cubic foot of liquid at 60 F expands to	850 cubic foot of gas
Reactivity	Easily absorbed by water
	Corrodes copper, zinc and their alloys.
	Compatible with iron, steel
	Highly reactive with mercury
	Incompatibility with polyisobutylenes, PVC and styrene copolymers
Major exposure hazards	Inhalation, skin contact, eyes contact, ingestion
Occupational exposure limits	OSHA PEL: 35 ppm

-28 F

10.3 HAZARDS

Hazards in ammonia refrigeration systems are associated with the chemical and physical characteristics of ammonia and the temperatures and pressures in the system. In general hazards for ammonia refrigeration systems can be classified in:

1. Hazards from the effect of low temperature: brittleness of materials at low temperatures. freezing of enclosed liquid. thermal stresses. changes of volume due to changes in temperature.
2. Hazards from excessive pressure caused by: increase in the pressure of condensation or pressure of saturated vapor. expansion of liquid refrigerant in a closed space without the presence of vapor and fire.
3. Hazards from direct effect of the liquid phase: excessive charge of the equipment presence of liquid in the compressors. liquid hammer in piping and loss of lubrication due to emulsification of oil.
4. Hazards from escape of refrigerants: fire. explosion. toxicity. freezing of skin. And asphyxiation.
5. Hazards from the moving parts of machinery: injuries. hearing loss from excessive noise. damage due to vibration and ignition of material due to broken pans.

6. Hazards from operation: excessive temperature at discharge. liquid slugging erroneous operation. reduction in mechanical strength caused
7. Hazards from corrosion. This category requires special consideration because the alternate frosting and defrosting of some parts of the system and the covering of equipment with insulation-

In order to help with the communication of the ammonia hazards. there are several safety classifications for ammonia . The National Fire Protection Association (NFPA) classifies ammonia as rating 3 for health hazards due to the corrosive effects on the skin. Rating 1 for fire hazards because it considers that it is difficult to ignite. And rating 0 for reactivity hazards because does not react violently with other substances.

Other classifications as ASHRAE. consider ammonia as "low flammability" because the heat of combustion is lower than 8174 Btu/lb and the LFL is above l4%. Also considers ammonia as “high toxicity" refrigerant because higher toxicity results from TLV are lower than 400 ppm.

Fire and explosion hazards of ammonia are presented in table 1.2. Ammonia is considered low flammability because in an outdoor situation, its flammability limits are difficult to reach. However, in confined spaces hazardous situations are possible and can cause fires and explosions.

Table 1.2 Fire and explosion hazards of ammonia [1]

Fire and Explosion Hazards	
Flash Point	N/A
Flammability limits	LFL 15 – 16%
	UFL 25 – 28%

Low peak pressures and slow rate of pressure rise are characteristic of ammonia explosions. Table 1.3 presents the explosion pressures for ammonia compared with pentane. Ammonia explosions are less violent and damaging than hydrocarbon explosions.

Table 1.3 Explosion pressures of ammonia and methane [1]

Explosion characteristic	Ammonia	Methane
Peak Pressure	~ 60 psig	~ 105 psig
Rate of Pressure Rise	440 psi/second	3000 psi/second

Ammonia fires are extinguished with water fog or spray, except if a pool of liquid ammonia is present. Fire extinguishing procedures include using water to mitigate vapours and vacate the area if concentration exceeds 5%.

Health hazards data of ammonia are inhalation, ingestion, skin contact and eye contact. Ammonia has an irritating odour that alerts of dangerous exposure. Odor threshold concentrations range from 1 ppm to 50 ppm. Nevertheless, acclimation occurs with chronic exposition to low concentrations of ammonia.

Effects of ammonia to health are severe because it is absorbed by the water in the tissues quickly.

Reactivity hazards are present if ammonia is in contact with strong acids, chlorine, bromine, mercury, silver and hypochlorites. Also, if temperature is higher of 600 F ammonia decomposes generating hydrogen.

Table 1.4 Ammonia concentrations and responses [1]

Concentration	Response
400 ppm	Immediate throat irritation
1,700 ppm	Cough
2,400 ppm	Threat to life after 30 minutes
>5,000 ppm	High likelihood of mortality with short exposure

Because all these hazards related to ammonia, several safety regulations were development for ammonia refrigeration system. They include the OSHA PSM 29 CFR part 1910 119 and EPA RPM 40 CFR part 68.Common elements for both regulation are hazard review, mechanical integrity, emergency response and operator training. EPA has guideline for OSHA and EPA regulation and also guidelines for equipment design and installation, operation, safety and operation procedures, start-up inspection and maintenance, water contamination, minimum safety criteria, room ventilation, machinery room design, identification of ammonia refrigeration piping and guidelines for avoiding component failure caused by abnormal pressure or shock.

10.4 NUIANCES AND RISKS ASSOCIATED WITH THE OPERATION OF INSTALLATIONS

A refrigeration installation like all other technical equipment, poses specific risks. These risks are especially related to the products used in the transfer loops to trap, transport and draw off excess calories. Generally speaking, the heat transfer fluid may be flammable, toxic and liable to effect the ozone layer in the upper atmosphere or participate in the "greenhouse effect”.

10.4.1 Nuisances

It should be reminded that refrigeration installations:

- Generally operate in a closed system and generate few nuisances during normal operation.
- Often are equipped with a cooling system using "evaporative" type refrigerant condensers possibly creating water vapour (steam) to a certain extent.
- Do not produce waste, excluding any used oil.

- Are equipped with potentially noisy equipment (compressor. Associated cooling equipment, etc.) and must be designed or equipped to limit noise pollution as much as possible.

10.4.2 Potential Risks

Fire/explosion risks

Ammonia is considered as a relatively non-flammable gas“- Its explosive limits in air are between 15 and 2896. However, a study indicates that the L.E-L. may be reduced by 4% in presence for a cloud consisting of oil (simultaneous lubricant leak) and aerosol ammonia.

The self-ignition temperature of ammonia is 630°C. As it dissolves in nitrogen beginning at 450 - 550°C , the combustion obtained can result from the hydrogen formed.

Although much greater than the majority of hydrocarbons, its minimal ignition energy (680 m1) is nevertheless less than that delivered by a switch spark.

Ammonia's flammable and explosive character, particularly in a confined space, is a subject of controversy. A bibliography compiled stipulates that all the flammability and explosively characteristics published indicate that ammonia is a combustible gas which is quite less reactive, vis-a-vis air than the majority of other combustible gas, and methane in particular. As such, the minimum ignition energy of an air ammonia mixture is greater. the flame in the mixture propagates with more difficulty and slower; the violence of the explosion is weaker in a closed recipient. The study sites a few accidents abroad in which ignition/explosion of ammonia is suspected.

10.5 TOXIC HAZARD

With the exception of air, rarely used in these conditions, all refrigerants are potentially harmful to man when their concentration in the air reaches a certain level. Fatal accidents resulting from anoxia have even been encountered with CFC .5. However, ammonia is one of the refrigerants whose toxicity is a dominant characteristic.

- Normally confined in the recipients and pipes of a refrigeration system ammonia can be released to the open air in an accidental situation especially resulting from:
- Normal operation of safety devices (valves. blow-out discs).
- Operational failure (a poorly-controlled purge of a circuit. etc).

- Through a limited leak (seal. loss of seal on a valve, corrosion etc).
- After equipment rupture (explosion caused by a fire. impact or equipment failure, etc.).

The ammonia released may then form a toxic cloud in the atmosphere and possibly cause water pollution if a permanent flow of water is located nearby (wastewater-"rainwater collector, etc.) or following inappropriate maintenance or servicing (sprayed water from a curtain not collected etc-).

- A limited leak corresponds to a continuous liquid or gaseous phase release and at a constant or nearly constant rate. Its duration depends on the technical characteristics of the installation the location of the "break". And the emergency response resources and the intervention time.
- The rupture instantly releases a significant quantity or all of the ammonia essentially in the form of an initial flash (up to 20943 of the mass of NH3 released for an ambient temperature of 25°C), generally followed by a second release corresponding to the slow vaporization of the residual liquid product released.

Chapter 11 OPERATING COST OF UNITS

11.1 Assumptions:

- Total operating hrs in year=8760
- Per unit cost of electricity= Rs 10/kWh

11.2 Compressor:

Initially with inlet conditions as -3℃ and 3.84 bar and outlet conditions as 97.3℃ and 14 bar. The power consumed and mass flow rate of refrigerant is given by:

$\dot{W}$=357.400kW

$\dot{m}$ref=1.295 kg/s

Units(kWh) of electricity consumed per annum=$\dot{W}$*operating-hrs

=357.4*8760

=31,30,824 kWh

Cost of electricity per annum= units consumed per annum* cost per unit

=3130824*10

=Rs 3,13,08,240

Now,

With inlet conditions as 0℃ and 3.84 bar and outlet conditions as 97.3℃ and 14 bar. The power consumed and mass flow rate of refrigerant is given by:

$\dot{W}$=302.000 kW

$\dot{m}$ref=1.280 kg/s

Units of electricity consumed per annum= $\dot{W}$*operating hrs

=302*8760

=26,45,520 kWh

Cost of electricity per annum= units consumed per annum* cost per unit

=2645520*10

=Rs 2,64,55,200

Cost saved per annum=31308240-26455200

=Rs 48,53,040

$\dot{W}$ (kW)	Operating hours	Unit consumed (kWh)	Rate of electricity (Rs./kWh)	Cost per annum (Rs.)	Cost Saved (Rs.)
357.400	8760	31, 30,824	10	3,13,08,240	**48, 53,040**
302.000	8760	26, 45,520	10	2,64,55,200	

11.3 Pumps:

There are four pumps to circulate water. Initially the mass flow rate of water is 56.100 kg/s. Now the flow rate is reduced to 42.105 kg/s.

Initially

η =0.750

P=15.000 kW

$\dot{m}$=56.100 kg/s

Ws=0.201 kj/kg

Now

η =0.750

$\dot{m}$= 42.105 kg/s

Ws=0.201 kj/kg

P=?

$$P = \frac{W * \dot{m}}{\eta} = \frac{0.201 * 42.105}{0.75}$$

=11.280 kW

ΔPower=15kW-11.28kW

=3.720 kW

For four pumps

Power saved =3.72 *4

=14.880 kW

Total operating hrs/annum= 8760

kWh-saved=14.880*8760

=130348.800 kWh

Cost saved =130348.800 kWh * Rs10/kWh

=Rs 1303488

Power (kW)	ΔPower (kW)	Power saved for 4 pumps (kW)	Operating hours per annum (hrs)	Units of electricity saved (kWh)	Rate of electricity (Rs./kWh)	Total cost saved per annum (Rs.)
15.000	3.720	14.880	8760	1,30,348.800	10	**13,03,488**
11.280						

11.4 Condenser:

Since the condenser is evaporative condenser, so two types of costs are involved:

1. For circulating air
2. For cooling water

11.4.1 For Circulating air:

For circulating air Fan is used which utilizes electricity for electric fan motor

Initially,

Power of fan motor= 37 kW

Air flow rate= 38.900 m^3/s

Assumption for $1m^3/s$ increase in flow rate, power increase is 0.951 kW

So, for now the Air flow rate=51.200 m^3/s

Increase in Power= 48.699-37.000

=11.699kW

Operating hrs per annum= 8760

Total units of electricity consumed= 8760*11.699

=102483.240 kWh

Increase in Cost/annum= 102483.240*10

=Rs.10,24,832.400

Power (kW)	ΔPower (kW)	Operating hours per annum (hrs)	Units of electricity increased (kWh)	Rate of electricity (Rs./kWh)	Total cost increased per annum (Rs.)
37.000	11.699	8760	102483.240	10	**10,24,832.400**
48.699					

11.4.2 For cooling water:

Increase in cost for pumping water:

η =0.750

P=4.000 kW

ṁ=36.600 kg/s

Ws=0.082 kJ/kg

Now

η =0.750

ṁ= 46.700 kg/s

Ws=0.201 kj/kg

P=?

$$P = \frac{\mathrm{W} * \dot{\mathrm{m}}}{\eta} = \frac{0.082 * 46.7}{0.75}$$

=5.099 kW

ΔPower=5.099 kW-4.000kW

=1.099 kW

Total operating hrs/annum= 8760

kWh increase =1.099*8760 =9632.846 kWh

Cost increase =9632.846 kWh * Rs10/kWh

=Rs 96328.460

Power (kW)	ΔPower (kW)	Operating hours per annum (hrs)	Units of electricity increased (kWh)	Rate of electricity (Rs./kWh)	Total cost increased per annum (Rs.)
4.000	1.099	8760	9632.846	10	**96328.460**
5.099					

11.4.3 Increase in cost of cooling water:

Price per litre of water= Rs. 0.01/Litre

Increase in flow rate=46.700-36.600

=10.100 kg/s

=31,85,13,600 kg/year

Increase in Cost per annum=0.01*318513600

=Rs 31,85,136

Flow rates (kg/s)	**ΔFlow rates (kg/s)**	**ΔFlow rates (kg/yr)**	**Operating hours per annum (hrs)**	**Rate of water (Rs./kg)**	**Total cost increased per annum (Rs.)**
46.700	10.100	31,85,13,600	8760	0.01	**31,85,136**
36.600					

Cost saved per annum (Rs.)	**Cost increase per annum (Rs.)**	**Net Cost Saved per annum (Rs.)**
Compressor=**48, 53,040**	Condenser pumps=**96328.460**	
Pumps=**13,03,488**	Cooling water= **31,85,136**	
	Fan motor=**10,24,832.400**	
Total=**61,56,528.00**	Total= **43,06,296.90**	Total= **18,50,231.14**

Chapter 12 CONCLUSION

The current COP of Engro's plant to produce chilled water is enhanced by using different design specifications and a new insight is presented of using R32 as a potential refrigerant and using VFDs (Variable frequency Drives).

The Evaporator and Condenser design calculations directly influence the cooling load and the compressor work. So the compressor is the same that is being used but will be run at lower power hence reducing electricity costs. An electricity cost of the compressor amount to around 40% of the refrigeration plant hence reducing its power is of prime priority.

Use of R32 as a suitable refrigerant will take a long way to be introduced into the industrial sector because of the wide use of ammonia but it is a step forward into the future. With its higher refrigeration capacity, use of smaller compressors R32 will be a potential research refrigerant.

VFD is a motor controller type that drives an electric motor by varying the frequency and the voltage supplied. Both of this can be used in the compressors and the air motors of the evaporative condenser. When we do not require the motor to run at full speed, VFD can help reduce the RPMs of that motor hence reducing the load and saving electricity.

All the different ways to increase the COP of the refrigeration system are studied and explained in detail. Further information can be found in the references listed.

Chapter 13 REFERENCES

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